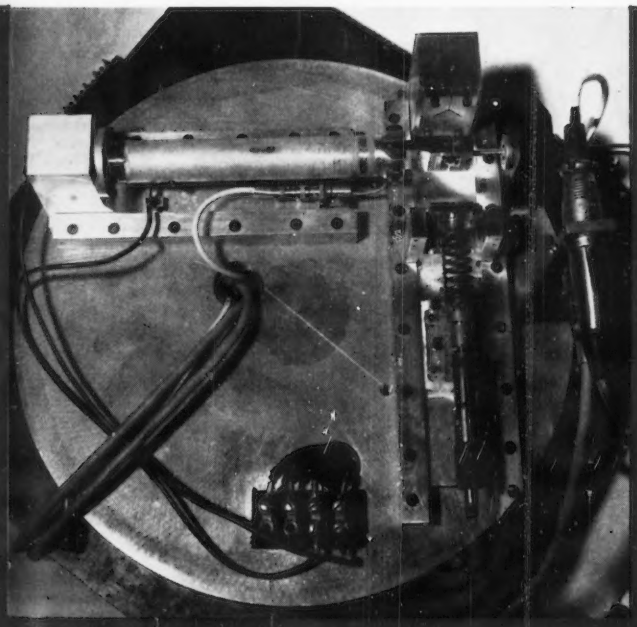
FEBRUARY 1961

experimental mechanics



SEE PAGE 48

IN THIS ISSUE

PAGE

33 Experimental Study of Large-diameter Thin-wall Pressure Vessels

43 Experiments in Mechanical and Optical Coincidence in Photoplasticity

48 Experiments on Damping at Contacts of a Sphere with Flat Plates

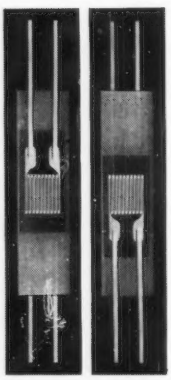
55 Measurement of Thermal Stresses in the S. S. Boulder Victory

SESA JOURNAL OF THE SOCIETY FOR EXPERIMENTAL STRESS ANALYSIS

NEW!
BUDD
FLEXAGAGE

(Pre-aligned double-deck pair of MetalFilm strain gage

For the first time, a single gage that measures both bending and axial strain



Mounts as a single, moisture-proof, ca brated gage *on one side of test surface.*

Eliminates difficult back-to-back ga mounting problems and need for pressur proof lead wire installation, such as in pre sure vessel analysis.

Makes possible bending strain analyses eve on surfaces formerly considered inaccessibl

Separates bending strains from direct (te sile, membrane, hoop) strains . . . accurate measures both.

*Patent Pend

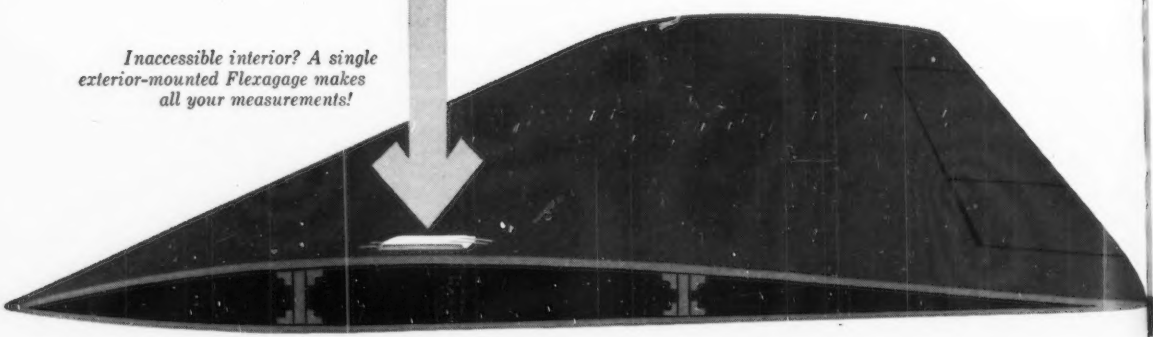
FOR FULL DETAILS, WRITE FOR NEW FLEXAGAGE BULLETI

INSTRUMENTS Budd DIVISION

THE BUDD COMPANY • P.O. Box 245 • Phoenixville, Pa

Consult your phone book for sales offices in: Atlanta, Ga.; Oak Park, Ill.; Detroit, Mich Seattle, Wash.; Dallas, Tex.; Los Angeles, Calif.
In Canada: Budd Instruments, Ltd., 170 Donway West, Don Mills, Ont.
Other Countries: Budd S.A., 10 Avenue de la Grande Armée, Paris 17e, France

Inaccessible interior? A single exterior-mounted Flexagage makes all your measurements!



Other gages from Budd:

- MetalFilm electrical strain gages in a wide range of sizes, configurations and temperature compensations.
- PhotoStress Rosettes—the only gages that provide directly both orientation and separate magnitudes of principal strains.
- PhotoStress gages for unidirectional strain measurements, transducer applications, machine alignment and torque measurements.

For details, circle No. 1 on Reader Information Card

experimental mechanics

VOLUME 1 • NUMBER 2 • FEBRUARY 1961

editor **B. E. Rossi**

asst. editor **C. H. Willer**

advertising **D. L. Vidmark**

art **Gil Miller**

OFFICERS AND EXECUTIVE COMMITTEE

D. C. Drucker *president*
I. Vigness *vice-president*
W. W. Soroka *vice-president*
W. M. Murray *treasurer*
B. E. Rossi *executive secretary*
F. C. Bailey
B. J. Lazan
J. C. New
C. E. Taylor
E. Wenk, Jr.

JOURNAL ADVISORY COMMITTEE

E. Wenk, Jr. *chairman*
D. C. Drucker
R. Guernsey, Jr.
B. J. Lazan
W. M. Murray
C. C. Perry
F. B. Stern, Jr.
I. Vigness

Published monthly by the Society for Experimental Stress Analysis. Publication office, 20th and Northampton Streets, Easton, Pa. Editorial and general offices, 21 Bridge Square, Westport, Conn. Subscription $10.00 per year in the United States and possessions; foreign countries $12.00 Single copies, nonmembers $1.50; members $1.00. Application to mail at second-class postage rates is pending at Easton, Pa.
Copyright 1961 by the Society for Experimental Stress Analysis. The Society is not responsible for any statement made or opinion expressed in its publications. Permission is given to reprint any article after its date of publicaton provided credit is given.

this month

TABLE OF CONTENTS

page

33 Experimental Study of Large-diameter Thin-wall Pressure Vessels
by A. J. Durelli, J. W. Dally and S. Morse

43 Experiments in Mechanical and Optical Coincidence in Photoplasticity
by M. M. Frocht and R. A. Thomson

48 Experiments on Damping at Contacts of a Sphere with Flat Plates
by L. E. Goodman and G. E. Bowie

55 Measurement of Thermal Stresses in the S. S. Boulder Victory
by J. L. Meriam, R. F. Steidel, G. W. Brown and P. T. Lyman

2A SESA Corporation Members

3A Experimentally Speaking

4A Tracking Station

6A New Products

14A New Literature

17A Reader Information Card

18A Index to Advertisers

19A SESA News

24A Section Reports

27A From the Campus

28A Personnel in the News

29A Reference Notebook

33A Membership Application Form

A Fast-growing List . . .

SESA CORPORATION MEMBERS

AEROJET-GENERAL CORP.
Sacramento, Calif.

ALLEGANY INSTRUMENT CO., INC.
Cumberland, Md.

ALUMINUM COMPANY OF AMERICA
Cleveland, Ohio

ARMCO STEEL CORP.
Middletown, Ohio

ARMOUR RESEARCH FOUNDATION
Chicago, Ill.

ASSOCIATION OF AMERICAN RAILROADS
Chicago, Ill.

AUTO SPECIALTIES MANUFACTURING CO.
Saint Joseph, Mich.

B & F INSTRUMENTS, INC.
Philadelphia, Pa.

THE BABCOCK & WILCOX CO.
Alliance, Ohio

BALDWIN-LIMA-HAMILTON CORP.
Waltham, Mass.

BELL TELEPHONE LABORATORIES, INC.
New York, N. Y.

BETHLEHEM STEEL CORP.
Bethlehem, Pa.

BREWER ENGINEERING LABORATORIES, INC.
Marion, Mass.

BROWN UNIVERSITY
Providence, R. I.

BUCYRUS-ERIE CO.
South Milwaukee, Wis.

THE BUDD CO. (AUTOMOTIVE DIVISION)
Detroit, Mich.

THE BUDD CO. (INSTRUMENTS DIVISION)
Phoenixville, Pa.

BYTREX CORP.
Newton, Mass.

CALIFORNIA INSTITUTE OF TECHNOLOGY
Pasadena, Calif.

CASE INSTITUTE OF TECHNOLOGY
Cleveland, Ohio

CATERPILLAR TRACTOR CO.
Peoria, Ill.

CHAIN BELT CO.
Milwaukee, Wis.

CHRYSLER CORP.
Detroit, Mich.

CONSOLIDATED WESTERN STEEL DIVISION (U. S. STEEL CORP.)
Los Angeles, Calif.

CONTINENTAL AVIATION & ENGINEERING CORP.
Detroit, Mich.

CONVAIR
San Diego, Calif.

CORNING GLASS WORKS
Corning, N. Y.

CUMMINS ENGINE CO.
Columbus, Ind.

CUSTOM SCIENTIFIC INSTRUMENTS, INC.
Kearny, N. J.

DATRAN ELECTRONICS
Manhattan Beach, Calif.

DIRECTION DE L'HYDRAULIQUE ET DE L'EQUIPEMENT RURAL
Alger, Algeria

DOMINION ENGINEERING WORKS, LTD.
Montreal, Canada

DOUGLAS AIRCRAFT CO., INC.
Santa Monica, Calif.

THE DOW CHEMICAL CO.
Midland, Mich.

E. I. DU PONT DE NEMOURS & CO., INC.
Wilmington, Del.

ELECTRO-MOTIVE DIVISION, GENERAL MOTORS CORP.
La Grange, Ill.

ELLIS ASSOCIATES
Pelham, N. Y.

FAIRCHILD ENGINE & AIRPLANE CORPORATION
Hagerstown, Md.

GENERAL ELECTRIC CO.
Schenectady, N. Y.

GILMORE INDUSTRIES, INC.
Media, Pa.

GRUMMAN AIRCRAFT ENGINEERING CORP.
Bethpage, L. I., N. Y.

HANDLEY PAGE, LTD.
London, N. W. 2, England

HAWKER AIRCRAFT LTD.
Kingston on-Thames, England

HIGH TEMPERATURE INSTRUMENTS CORP.
Philadelphia, Pa.

IMPERIAL CHEMICAL INDUSTRIES, LTD.
Billingham, England

INGERSOLL-RAND CO.
Phillipsburg, N. J.

INSTRON ENGINEERING CO.
Canton, Mass.

INTERNATIONAL BUSINESS MACHINES CORP.
Poughkeepsie, N. Y.

INTERNATIONAL HARVESTER CO.
Chicago, Ill.

KELSEY-HAYES CO.
Detroit, Mich.

KINETICS CORP.
Cincinnati, Ohio

KOEHRING COMPANY TEST AREA
Oconomowoc, Wis.

THE KORFUND CO., INC.
Long Island City, N. Y.

LEEDS & NORTHRUP CO.
North Wales, Pa.

LESSELLS & ASSOCIATES, INC.
Boston, Mass.

LINK-BELT CO.
Indianapolis, Ind.

LOCKHEED AIRCRAFT CORP., MISSILES & SPACE DIVISION
Sunnyvale, Calif.

LYCOMING DIVISION, AVCO MANUFACTURING CORP.
Stratford, Conn.

MAGNAFLUX CORP.
Chicago, Ill.

THE MARTIN CO.
Baltimore, Md.

MASSACHUSETTS INSTITUTE OF TECHNOLOGY
Cambridge, Mass.

MCDONNELL AIRCRAFT CORP.
St. Louis, Mo.

MICRO SYSTEMS, INC.
San Gabriel, Calif.

THE NATIONAL SUPPLY CO.
Pittsburgh, Pa.

NEWPORT NEWS SHIPBUILDING & DRY DOCK CO.
Newport News, Va.

NORTH AMERICAN AVIATION, INC.
Los Angeles, Calif.

ORMOND INSTRUMENTATION CENTER
Santa Fe Springs, Calif.

EDWARD A. OSSMANN & ASSOCIATES, INC.
Rochester, N. Y.

PANORAMIC RADIO PRODUCTS, INC.
Mount Vernon, N. Y.

PITTSBURGH PLATE GLASS CO.
Harmarville, Pa.

POLARIZING INSTRUMENTS CO., INC.
Irvington-on-Hudson, N. Y.

PURDUE UNIVERSITY, SCHOOL OF ENGINEERING
Lafayette, Ind.

RADIO CORPORATION OF AMERICA
Moorestown, N. J.

REGIE NATIONALE DES USINES RENAULT
Billancourt, Seine, France

REMINGTON RAND UNIVAC, DIVISION OF SPERRY RAND CORP.
St. Paul, Minn.

RENSSELAER POLYTECHNIC INSTITUTE
Troy, N. Y.

RESEARCH, INC.
Hopkins, Minn.

RESEARCH LABORATORIES, GENERAL MOTORS CORP.
Warren, Mich.

REVERE CORPORATION OF AMERICA
Wallingford, Connecticut

STATHAM INSTRUMENTS, INC.
Los Angeles, Calif.

STEIN ENGINEERING SERVICES, INC.
Phoenix, Ariz.

SUPERIOR STEEL & MALLEABLE CASTINGS CO.
Benton Harbor, Mich.

SVERDRUP & PARCEL, INC.
St. Louis, Mo.

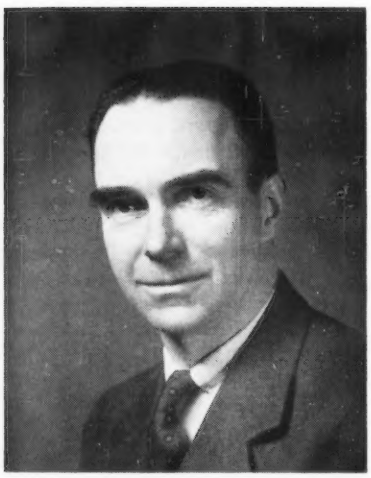
TRACTOR & IMPLEMENT DIVISION, FORD MOTOR CO.
Birmingham, Mich.

UNIVERSITY OF MINNESOTA
Minneapolis, Minn.

WESTINGHOUSE ELECTRIC CORP.
East Pittsburgh, Pa.

CORPORATE MEMBERSHIP means multiple benefits! For complete details write to Society for Experimental Stress Analysis, 21 Bridge Square, Westport, Conn.

Experimentally Speaking



THE PRICE OF KNOWLEDGE

Wherever we turn, we are reminded of the high cost of education. However, it seems that very few of us stop to consider the alternative aspect, "How much does it cost to be without knowledge?" This is a much more difficult question to answer in terms of dollars, but a moment's reflection will yield some interesting conjectures.

For example, if we knew how to control the destructive violence of storms, and had worked out means whereby the energy, which now produces harmful effects, could be applied to useful purposes, untold billions could be saved in terms of loss of life, property damage, and injury to agricultural crops, to say nothing of the value of the energy which might be made available. Additional benefits would also accrue by directing the time and effort, which are now expended in repairing storm damage, to gainful production. We can all appreciate that the cost of our lack of knowledge of storm control runs to a very high figure annually. Many other examples can be cited, but this one should be sufficient to suggest the cost of ignorance.

Who is to pay for acquiring, and handing on, all the growing mass of knowledge which is to be made available in the future? This is an intriguing question, but it seems reasonable to assume that both the information itself, and the cost of producing it, will have to be contributed from an infinitely large number of comparatively small sources. The real problems, however, will be to develop talent and to store the information conveniently and accessibly. Good teachers—using the term in its general, rather than classroom sense—are needed to transmit knowledge and to stimulate others to add to it as they hand it on.

In the specialized areas of experimental mechanics, parallel situations exist. The expense of developing knowledge may seem high. Yet the cost of ignorance may not be fully appreciated until some spectacular disaster draws the situation into sharp focus, and subsequent investigation brings forth the knowledge which could have prevented the misfortune.

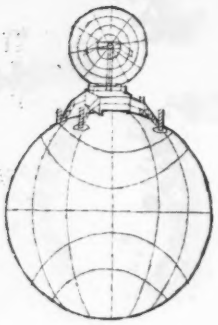
The SESA has been fortunate in growing in an atmosphere of free interchange of information. A cooperative spirit and a willingness to share the fruits of research have prevailed since the early Photoelasticity Conferences which preceded it. As a result of this, the Society now has an international membership, and enjoys local, national, and international meetings. Numerous technical contributions have been received from abroad and the *Proceedings* have been distributed in many countries.

In its field, the Society has long been noted for educational lectures for beginners. More recently, programs dealing with both fundamentals and advanced techniques have been presented under university sponsorship.

With the introduction of EXPERIMENTAL MECHANICS, in January of this year, the Society has taken one of its greatest steps in the direction of promoting and disseminating knowledge.

Let us continue *full speed ahead!*

William M. Murray
TREASURER, SESA



tracking station

the industry in focus

Technical Advisory Committees Established by NBS

Two new Technical Advisory Committees have been established by the National Bureau of Standards, U. S. Department of Commerce, one on Calibration and Measurement Services, and one on Engineering and Related Standards.

The committees include leaders in specialized fields, drawn from industry. The purpose of the committees is to aid the Bureau in cooperating with industry in the fields of precision measurement, calibration and standard practices.

William Wildhack, Special Assistant to the NBS Director, is chairman of both committees.

The Advisory Committee on Calibration and Measurement Services is an outgrowth of the tremendous increase in measurement and calibration activity in private industry and the military services in the last two years. The "space-age" industries have led this growth, along with computer, microwave component and electronic industries. Numerous studies and surveys conducted by industry and the military show pressing needs for expanded calibration facilities, improved standards and more and better measuring equipment.

Program Announced for 1961 Temperature Symposium

The technical program is now complete for the most comprehensive meeting ever held on the subject of temperature measurement. About 200 papers will cover temperature measurements from absolute zero to 10,000,000° K with interests ranging from the temperature of the human body to the temperature of the sun. Almost every area of the physical, biological and medical sciences will be represented at the 1961 Symposium on "Temperature, Its Measurement and Control in Science and Industry," to be held in Columbus, Ohio, Mar. 27–31, 1961. The conference is sponsored by the American Institute of Physics, the Instrument Society of America and the National Bureau of Standards.

Regional Sales Offices Opened by NLS

Non-Linear Systems, Inc., of Del Mar, Calif., has announced the opening of four regional sales offices and the appointment of a sales manager for each office. Regional offices are in Garden City, N. Y.; Chicago, Ill.; Washington, D. C.; and South Pasadena, Calif. James F. Helfrich has been named sales manager for the northeastern marketing region, with offices at 1000 Franklin Ave., Garden City, N. Y.

Thomas Kurtzer has been named sales manager for the central marketing region, with offices at 1915 N. Harlem Ave., Chicago, Ill.

W. E. Bradbury, SESA member, has been named sales manager for the Eastern Marketing Region, with offices at 1925 K St., N. W., Washington, D. C.

William Olbrich has been named sales manager for the Western Marketing region, with offices at 1910 Fair Oaks, South Pasadena, Calif.

Magnaflux Acquires Metal Control Laboratories

Magnaflux Corp., Chicago, a wholly owned subsidiary of General Mills has acquired Metal Control Laboratories, Los Angeles, to effect a complete complement of integrated nondestructive, metallurgical and physical testing facilities and skills. The acquisition was a cash transaction.

R. A. Wilson, Magnaflux president and vice-president of General Mills, said in making the recent announcement that Metal Control Laboratories will be operated as a part of the Magnaflux organization, working closely with the present Magnaflux Materials Testing Laboratories at Los Angeles and Van Nuys.

Statham Development Merges with Statham Instruments

Statham Development Corp., a wholly owned subsidiary of Statham Instruments, Inc., has been merged with the parent company. Under the new arrangement, what has been known as Statham Development Corp. is now the Electronics Division of Statham Instruments, Inc., Los Angeles, Calif.

The purpose of this merger is to afford a more efficient utilization of the parent company's corporate facilities.

It was also announced that Earl W. Grant has been elected by the board of directors of Statham Instruments, Inc., as vice-president and manager of the Electronics Division. Mr. Grant was formerly vice-president and general manager of Statham Development Corp.

Marquardt Reorganizes Power Systems Group

The Marquardt Corp. of Van Nuys, Calif., has announced organizational changes in its power systems group, primarily devised to integrate the group's sales activities with its divisional engineering functions. Concurrent with the reorganization is the establishment of a new special engines division and a combined manufacturing and material division.

The power systems group consists of five engineering divisions: propulsion, controls and accessories, nuclear systems, facilities engineering, and the new special engines division. In addition three divisions serve as support functions for the power systems group: test, administration and controls, and the new manufacturing and material division.

Report Highs in Sales

Beckman Instruments, Inc., Fullerton, Calif., has reported new first-quarter highs in sales and operating earnings for the period ending Sept. 30, 1960. Preliminary figures indicate sales of approximately $15 million, compared with $12,638,315 for the corresponding period a year ago. Operating earnings should range from $700,000 to $775,000 or 50 to 55 cents a share, compared with 1960 first-quarter earnings of $622,516 or 46 cents a share.

FUTURE MEETINGS

SESA

MAY 10–12. Spring Meeting. Benjamin Franklin Hotel, Philadelphia, Pa.
NOVEMBER 1–3. First International Congress on Experimental Mechanics. Hotel New Yorker, New York, N. Y.

• Related Conference:
OCTOBER 29–31. International Symposium on Photoelasticity. Illinois Institute of Technology, Chicago, Ill.

ARS

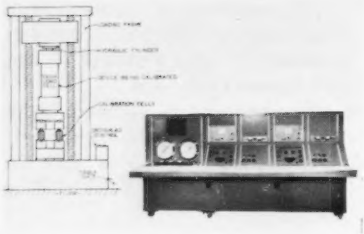
APRIL 5–7. Conference of the American Rocket Society. Lifting Re-entry Vehicles: Structures, Materials, Design. Palm Springs, Calif.

ASTM

JUNE 25–30. Annual Meetings of American Society for Testing Materials. Chalfonte-Haddon Hall, Atlantic City, N. J.

ISA

APRIL 17–19. Symposium on Instrumental Methods of Analysis. Instrument Society of America. Shamrock Hilton Hotel, Houston, Tex.



LOAD-CELL CALIBRATION SYSTEM

Details on their new Model 256 load-cell calibration system for missiles have just been announced by Gilmore Industries, Inc., 13015 Woodland Ave., Cleveland 20, Ohio.

According to the manufacturer, the Model 256 is a highly accurate instrument for calibrating load-cell transducers.

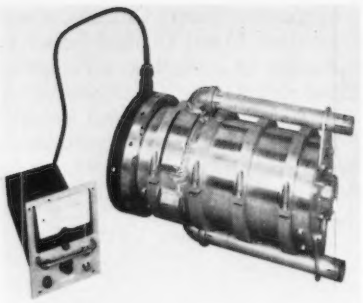
The system calibrates from 5000 to 1,000,000 lb within 0.05% of reading by using three overlapping sets of Gilmore M-170 force calibrators. The press is hydraulically operated and arranged to handle tension and compression loads, with a stroke opening from 0 to 96 in.

For details, circle No. 51 on Reader Information Card.

REACTION TORQUEMETER

Bytrex Corp., 50 Hunt St., Newton 58, Mass., announces the availability of the Series PT reaction torquemeters for use with water brakes (absorption dynamometers) and other auxiliary-driven, power-absorbing devices, such as pumps, generators, compressors, etc.

The manufacturer states that the



Series PT torquemeters are compact, highly reliable and accurate devices which measure the torque transmitted from the prime mover to the power receiver without the use of rotating or moving parts. The pickup is installed in the same manner as a simple adaptor flange and is available in standard sizes from 200 to 20,000 in. lb.

The Bytrex PT Series torquemeters are available with a variety of readouts for indication, recording, controlling, etc.

For details, circle No. 52 on Reader Information Card.

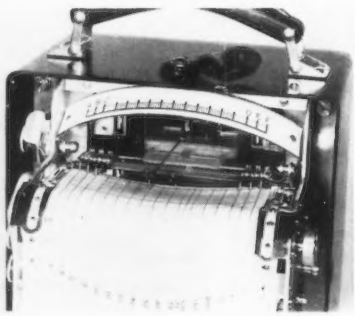
CURVE FOLLOWER

It is only necessary to plot a repetitive function variation once with the new Data-Trak curve follower offered by Research, Inc., Box 6164, Minneapolis 24, Minn. Continuous rotation of the data drums in the new models produces cyclical output signal variations from a pencil drawn "pattern," according to the manufacturer.

Model FGE 5048 Data-Trak is said to interpret function curves drawn

as a double line with ordinary pencil on common graph paper. No special paper, pencils or conductive inks are required. The penciled chart, mounted on Data-Trak's revolving drum, guides a servo-driven capacitive probe that follows the centerline between the plotted lines. A potentiometer, geared to the probe drive mechanism, divides any impressed voltage in precisely the same proportion as the drawn curve divides the graph scale. Since the probe does not actually touch the paper, no chart wear occurs and the charts can be stored and reused indefinitely.

For details, circle No. 53 on Reader Information Card.



new products

DATA-RECORDING SYSTEM

Datex Corp., 1307 S. Myrtle Ave., Monrovia, Calif., reports that a simple, reliable and economical method for recording, in digital form, the output of strain gages is provided in the Datex data-recording system. The unit gives stress data in immediately usable pounds-per-square-inch figures, and is particularly useful in mass production testing of similar units. The data are recorded on paper tape by a Datex-Monroe data/log printer at a rate of 0.7 to 5 sec per point.

For details, circle No. 55 on Reader Information Card.

ANALOG-EVENTS RECORDER

The Esterline-Angus Instrument Co., P. O. Box 596, Indianapolis 6, Ind., announces a new recorder called the analog-events recorder.

The center portion of the chart (3 in.) is an analog measurement usually made by a basic permanent-magnet moving-coil movement which by internal circuitry may be made to measure d-c microamperes, d-c milliamperes, d-c millivolts, d-c volts, a-c volts and a-c amps. Also speed, pressure and many other quantities can be measured through the use of suitable transducers.

Six chronograph pens or event markers showing on-off time of related events or simply timing pulses can be synchronized with the analog record by marking in the enlarged margins of the same chart.

For details, circle No. 54 on Reader Information Card.

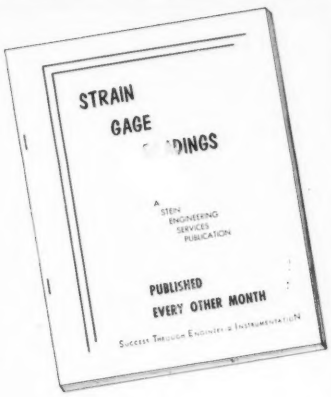
Sources of information on

STRAIN GAGES

STRAIN GAGE TECHNIQUES

by W. M. Murray
& P. K. Stein

A comprehensive reference book on basic theory and experiments

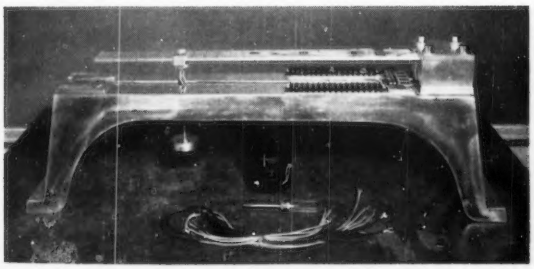
669 pages, 8½ x 11, cloth bound, photo-offset from typewritten copy

STRAIN GAGE READINGS

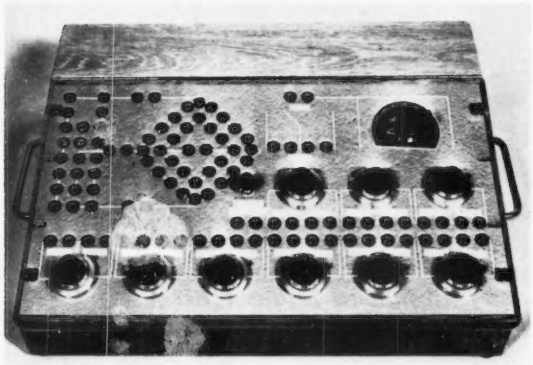
P. K. Stein, Editor

A bi-monthly journal devoted to new applications and techniques and reviews of current literature

THE STRAIN GAGE INSTRUCTOR

Aluminum stand and cantilever beam with mounted strain gages





Plug board for strain gage & instrumentation circuits

EXPERIMENTAL EQUIPMENT

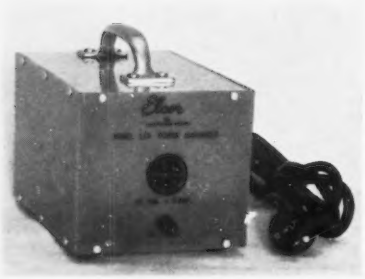
for demonstrating the theory and performing the laboratory experiments which are described in STRAIN GAGE TECHNIQUES and STRAIN GAGE READINGS
Available from

Stein Engineering Services, Inc.

5602 East Monte Rosa, Phoenix, Arizona

Write for prices and particulars

For details, circle No. 11 on Reader Information Card

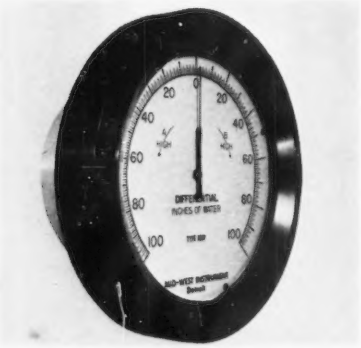


POWER ISOFORMER

Elcor, Inc., Falls Church, Va., announce the availability of Model L131 power isoformer, a 350-w isolation transformer featuring low distributed capacitance between the secondary winding and ground. Intended for general-purpose laboratory use or for mounting as a component in equipment, this transformer permits isolating from ground (i.e., "floating") a complete instrument or a component circuit, according to the manufacturer. The isolation from the power line and ground is said to be so effective that signals from direct current to several megacycles can be applied between the chassis of a typical conventional instrument and ground in applications where such operation is necessary or desirable because of interconnection with other equipment.

The Power Isoformer may be employed either for isolating a complete instrument or for isolating a component circuit.

For details, circle No. 58 on Reader Information Card.



DIFFERENTIAL GAGE

Mid-West Instrument, Division of Astra Associates, Inc., 1021 E. State Fair, Detroit 3, Mich., announces that their new type 109 differential pressure gage covers working pressures to 4000 psi at 100 in. H_2O to 400 psid. Encapsuled bourdon-tube element provides greater overrange protection and reduces maintenance costs. New jeweled bearing movement assures magnetic transmission of bourdon tube motion—with extremely low hysteresis and tracking error, according to manufacturer. Dial sizes are $4^1/_2$ and 6 in.

For details, circle No. 60 on Reader Information Card.

X-RAY UNITS

Two new Norelco industrial X-ray units (PG 140 and PG 200), described as compactly designed for minimum weight and to provide high penetrating power and excellent radiographic definition, are announced by Philips Electronic Instruments, 750 S. Fulton Ave., Mount Vernon, N. Y.

Completely self-contained, the new units are shockproofed, rayproofed and weatherproofed for outdoor operation, the manufacturer states. The two tube heads (140 and 200 kv, respectively) are companion units and one control cabinet serves either head.

The PG 140 tube head weighs 55 lb and the PG 200 tube head weighs 88 lb. Lightweight cables are employed for connection to the control cabinet. Tube heads can be hoisted to remote areas for radiography on aircraft wing sections, welds and steel components up to $2^1/_2$ in. in thickness.

For details, circle No. 62 on Reader Information Card.

HIGH-TEMPERATURE INDUCTION MOTOR

Kearfott Division of General Precision, Inc., 1150 McBride Ave.,

PHOTOELASTIC STRESS ANALYSIS WITH POLAROID FILTERS*

Standard Equipment or Special

Complete Apparatus or Components

Point Light Sources
White Light and Monochromatic (546 or 589 line)

Matched Quarter Wave Plates to 17″ diameter

Polaroid Polarizers to 17″ diameter
Cemented Glass Laminations #6/006

Straining Frames

Recording Cameras

* *T. M. Reg. U. S. Pat. Off. by Polaroid Corporation*

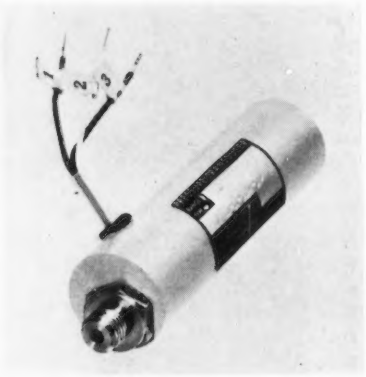
THE POLARIZING INSTRUMENT COMPANY, Inc.
IRVINGTON-ON-HUDSON, NEW YORK

For details, circle No. 5 on Reader Information Card

Little Falls, N. J., has announced the addition of the A-10-11 unit to its line of miniature and subminiature special service motors. These special service motors are manufactured by the Hertner Electric Co., a wholly owned subsidiary of Kearfott. This motor incorporates the feature of operating continuously over an ambient temperature range of −55° to +125° C.

Totally enclosed, this extremely compact motor is also specifically designed to exhibit low starting torque.

For details, circle No. 63 on Reader Information Card.



SMALL-DIAMETER HIGH-PRESSURE TRANSDUCERS

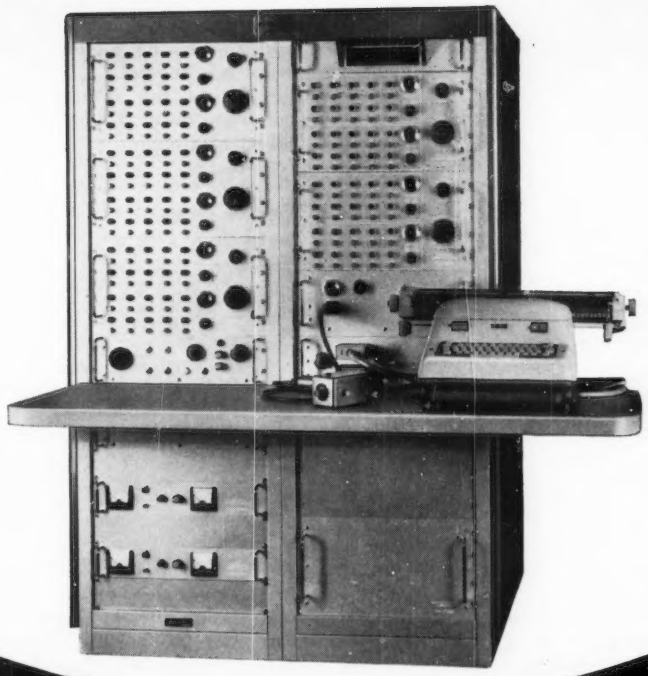
Development of miniature, high-pressure, high-temperature pressure transducers, capable of measuring up to 15,000 psi and weighing less than 0.25 lb, was announced recently by Colvin Laboratories, Inc., 364 Glenwood Ave., East Orange, N. J.

Designated as Series 8416, they are believed to be the smallest-diameter transducers for measurement of high-pressure ranges. The units measure only 1 in. in diameter by 3.65 in. long.

The manufacturer reports that the transducers represent a reduction in the previous size of such devices of approximately 50%. They are designed to measure pressures in any range from 0 to 400 psi to 0 to 15,000 psi and are available in various resistance values. The new transducers are an exact complement of Colvin Laboratories' Series 401 miniature transducers which are limited to pressure ranges from 0–1 to 0–400 psi.

The transducers can be manufactured to measure gage, absolute or differential pressures. For special applications, they can be made to withstand temperatures up to 600° F, vibration as high as 35 g's and operation in corrosive atmospheres such as inhibited red fuming nitric acid or other harmful materials.

For details, circle No. 56 on Reader Information Card.



NEW... VERSATILE DATA LOGGER FOR STRAIN GAGE AND MILLIVOLT INPUTS

- ☐ 1, 2 and 4 arm bridges with gage resistances from 60-1000 ohms may be used interchangeably in all channels.
- ☐ Millivolt inputs anywhere in the range from 10-100 resolved to 1 part in 5000.
- ☐ 100 channels expandable to 200 channels by the addition of standard 20 channel modules.
- ☐ Error indication for shorted or grounded gages with complete channel isolation to prevent affecting channels.
- ☐ Separable input panels, 5 terminal hook up configuration and automatic zero balance reduce operator fatigue, minimize test time and provide maximum equipment utilization.

Bytrex equipment is designed and built by people who understand and use strain gages. Our Data Loggers will meet your highest performance standards. *Please write for Bulletin 109.*

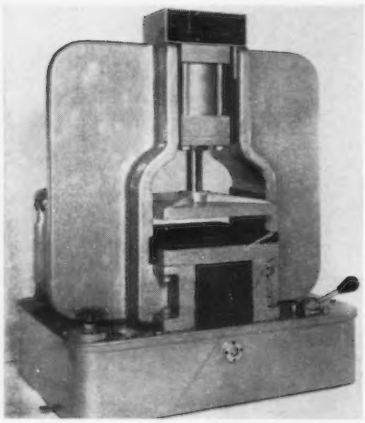
BYTREX®

BYTREX CORPORATION

50 HUNT ST. • NEWTON 58, MASSACHUSETTS

Designers and manufacturers of strain indicators, switching and balancing units, strain gage amplifiers, precision bridges, semiconductor gages and devices, special purpose measuring systems for weight, force, pressure and torsion.

For details, circle No. 2 on Reader Information Card



LABORATORY SAMPLE CUTTER

Thwing-Albert Instrument Co., Penn St. and Pulaski Ave., Philadelphia 44, Pa., announces the development of a new electro-hydraulically operated laboratory sample cutter.

The Model No. 240-4 Alfa laboratory sample cutter, more powerful than hand-operated models, is adjustable to apply up to 8 tons of pressure at the cutting surface, according to the manufacturer.

It is also reported that finger-tip operation makes it possible to cut larger, thicker and harder laboratory samples of paper, plastic, rubber, etc.

For details, circle No. 74 on Reader Information Card.

DIGITAL COMPUTER

General specifications on Recomp II, a general purpose, all-transistor, single-address, digital computer are available from North American Aviation's Autometrics Division, 3584 Wilshire Blvd., Los Angeles 5, Calif. According to the manufacturer, this computer is capable of solving problems of extreme complexity with split-second rapidity. It comes equipped with built-in floating point arithmetic and automatic decimal conversion. Its transistor-compact construction is said to provide mobility from one location to another.

It has five components: computer and memory, photoelectric tape reader, typewriter, tape punch and console.

For details, circle No. 75 on Reader Information Card.

DATA-TRANSLATION EQUIPMENT

The Systems Division of Beckman Instruments, Inc., 325 N. Muller Ave., Anaheim, Calif., announces the extension of its line of Model 210 data systems into the field of data translation.

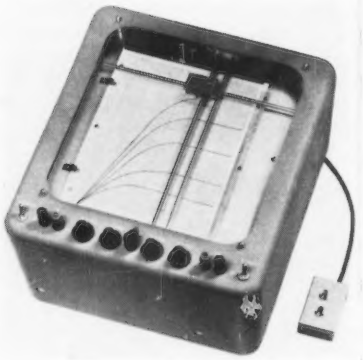
Available in a wide variety of input-to-output combinations, all Beckman data translators utilize the Model 210 system as the basic translator. Functional modules, in the form of direct-coupled logic cards, are incorporated into the system to tailor it to a particular application. Use of the time-tested Model 210 substantially reduces engineering and design time, and increases the system's flexibility, according to the manufacturer. All-solid-state modules are easily replaced in the field, permitting system modification as requirements change.

For details, circle No. 76 on Reader Information Card.

ELECTRIC PEN-LIFTER XY RECORDER

The Houston Instrument Corp., P. O. Box 22234, Houston 27, Tex., announces that their new Model HR-93 XY recorder has a simple and rugged type of electric pen-lift mechanism allowing point plotting, family curve tracing and rapid nonrecording pen indexing. The pen lifter can either be operated with the hand-held manual control box or can be tied into the test circuitry for automatic operation.

You'll find
READER INFORMATION CARD
on Page 17A



The manufacturer explains that a load-operate switch automatically picks up the pen and indexes it away from the chart area so that the graph paper may be loaded easily with no trace appearing on the chart. Further, after loading, the pen is held up so that no trace appears when indexing back to the original position.

Accuracy is said to be $\frac{1}{2}\%$ with $7\frac{1}{2}$ ips pen speed on both axes. Snap on pen construction allows a quick change for various color traces.

For details, circle No. 77 on Reader Information Card.

HARGRAVE CLAMPS — CHISELS — PUNCHES — MASONRY DRILLS

PRO-TECTO-HED PREVENTS MUSHROOMING

FULL RANGE PATTERNS & SIZES

WRITE FOR FREE
GENERAL CATALOG
THE CINCINNATI TOOL CO.
4356 Montgomery Rd.
Cincinnati 12, Ohio

CARBIDE TIPPED MASONRY DRILLS

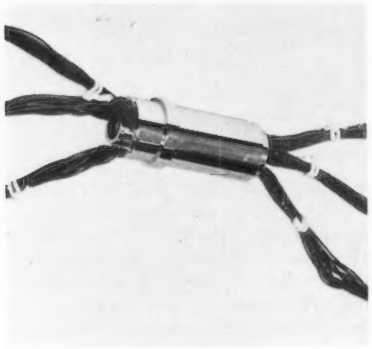
For details, circle No. 4 on Reader Information Card

"PORTABLE" TEST CHAMBER

Statham Instruments, Inc., 12401 W. Olympic Blvd., Los Angeles 64, Calif., announce the introduction of what is described to be the largest "portable" temperature test chamber available, with a testing area of 8 cu ft. Designated the "MC" for module chamber, this unit operates from $-100°$ to $+750°$ F with accuracy to $\pm 5°$ F. The heating and cooling rates are variable from 5° F per minute, depending upon the test load.

According to the manufacturer, modular design of the MC makes it suitable for a wide variety of applications, including laboratory, production line, quality control, life testing and plastics curing.

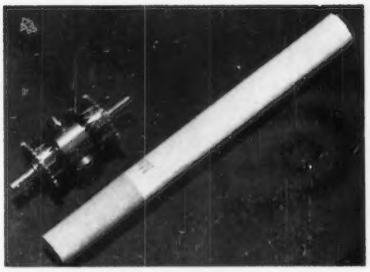
For details, circle No. 70 on Reader Information Card.



SLIP RINGS

A slip-ring assembly, specifically designed to withstand extremely high vacuums and the intense radiation of outer space conditions, has been designed and built by Slip Ring Company of America, 3612 W. Jefferson Blvd., Los Angeles 16, Calif. Designated as A72-ring assembly, it contains both power and signal circuits.

For deatils, circle No. 71 on Reader Information Card.



MINIATURE DIFFERENTIAL

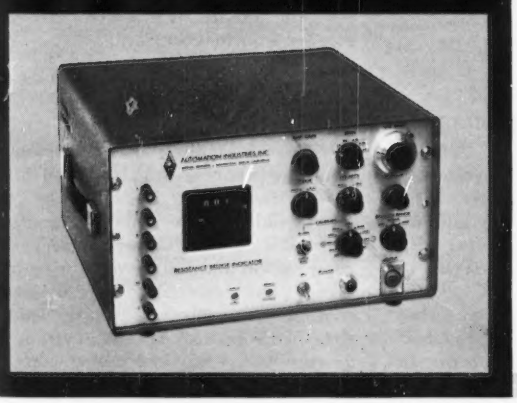
Ford Instrument Co. Division of Sperry Rand Corp., 31-10 Thomson Ave., Long Island City 1, N. Y., announces the addition of a $^3/_{32}$-in. standard differential to its line of single spider-gear differentials. It provides extreme accuracy in additive and subtractive operations while functioning with minimum friction and backlash, according to the manufacturer.

The $^3/_{32}$-in. differential operates in a working circle of 0.585 in. maximum diameter and weighs less than $^1/_2$ oz with typical end gears. The breakaway torque is as low as 0.07 oz-in., assuring high sensitivity. Maximum lost motion does not exceed 20 min of arc. Stainless steel is used throughout.

For details, circle No. 72 on Reader Information Card.

RESISTANCE BRIDGE INDICATOR

RBI



The Datran Resistance Bridge Indicator is a self balancing, accurate, and compact instrument for measuring outputs of resistance bridge transducers in the range of 50 to 1000 ohms. Output from either full or half bridges with 1, 2, or 4 active arms are read from a digital display without necessity for external accessory equipment or manual null-balance operation. The "RBI" is calibrated to read directly in microinches per inch strain with strain gage inputs. Variable span control permits transducers such as load, deflection, and pressure pickups to be calibrated directly in terms of the quantities being measured (i.e. pounds, inches, psi or any desired units). The unique servo balancing circuits have been developed specifically for resistance bridge instrumentation. Unlike other methods, calibration is determined entirely by precision resistance networks, essentially independent of the value of bridge excitation voltage. Input impedance of the indicator is 250,000 ohms. A single bridge can be operated into two or more similar instruments simultaneously, with negligible loading and interaction effects. An adjustable direct voltage supply furnishes necessary bridge power. Since accuracy is independent of supply voltage variations, there is no need for batteries, standard cells, or standardization.

For a more comprehensive description please address inquiries to:

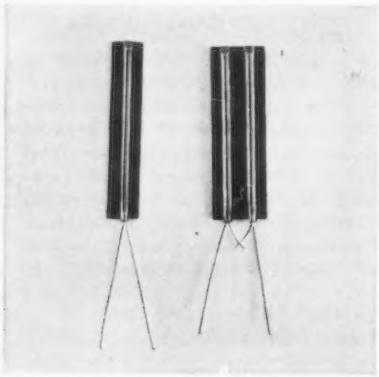
AUTOMATION INDUSTRIES, INC.
DATRAN DIVISION • 7750 Monroe St., Paramount, Calif.

For details, circle No. 13 on Reader Information Card

Microdot Acquires Micro-Test

Los Angeles — Microdot Inc. has entered the field of strain measurement devices through the acquisition of Micro-Test, Inc. of Santa Monica, California. Through this move, Microdot will manufacture and market the only weldable strain gage in the U.S. — a device that is capable of continuous operation at 800°F and dynamic test to 1500°F.

The weldable strain gage, used extensively in missile, space, and atomic projects, will be integrated with Microdot's existing pressure, force, and acceleration measurement capabilities.



Weldable strain gages, one-piece, etched wire filaments in swaged stainless steel tubes, are suitable for use from cryogenic to elevated temperatures. Photo shows typical gages in actual size.

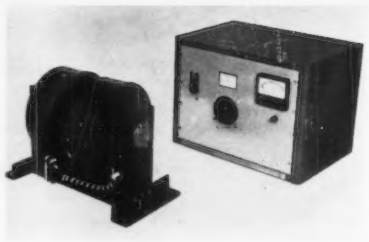
The Micro-Test acquisition is in line with other current moves by Microdot to enhance and broaden the company's instrumentation capabilities. Other Micro-Test products to be offered by Microdot include: temperature sensors in 60 and 120 ohm resistances that are completely insensitive to strain, and hermetically sealed load cells with adjustable temperature sensitivity offered in ranges of from 5,000 to 50,000 pounds for both compression and tension-compression.

Microdot sees great potential for the application of these new devices wherever physical phenomena can be measured and controlled. The weldable strain gage is already being used in atomic reactors, the exterior of submarines, inside chemical vats, on pilings sunk into the St. Lawrence Seaway, and a wide variety of airborne defense projects.

MICRODOT INC.

220 Pasadena Ave.
South Pasadena, California

For details, circle No. 3 on Reader Information Card



LABORATORY MAGNET SYSTEM

MHD Research, Inc., 1571 Placentia Ave., Newport Beach, Calif., announces the production of a line of versatile laboratory magnet systems.

These flexible general-purpose magnet assemblies are described as being ideal for a variety of applications. Individually adjustable and replaceable poles and pole faces allow maximum variation of the magnetic-field configuration.

Copper-wound coils are insulated with Class B materials and are calculated for continuous duty without additional cooling.

Performance specifications are available upon request.

For details, circle No. 64 on Reader Information Card.

ACCELEROMETER-AMPLIFIER SYSTEM

The development of miniature, completely solid-state accelerometer-amplifier system for missile and airborn shock and vibration measurements, packaged in a single unit, has just been announced by Columbia Research Laboratories, MacDade Blvd. & Bullens Lane, Woodlyne, Pa. The system is ideal for applications where voltage levels analogous to shock and vibration data must have sufficient amplitude to modulate subcarrier oscillators in a telemetering system.

Designated as the Model 50X2 accelerometer-amplifier system, the device contains a true compression-type accelerometer operating on the piezo-electric effect and a transistorized high-gain amplifier with a broad frequency response.

For details, circle No. 65 on Reader Information Card.

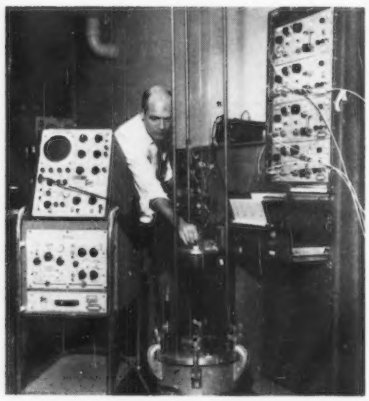
TRANSISTORIZED NULL INDICATOR

Gertsch Products, Inc., 3211 S. La Cienega Blvd., Los Angeles 16, Calif., announces a small-size, fully transistorized phase-sensitive null indicator. Unit may be combined with any standard Gertsch RatioTran to form an a-c ratio bridge.

The manufacturer states that three ranges of sensitivity are provided: 10μv, 100μv and 1000μv. A maximum sensitivity of 10 microvolts allows excellent resolution for bridge-circuit operation. Frequency range is 50 cps to 10 kc.

Designated as Model NI-2, unit has a relatively high input impedance of one megohm due to "boot strapped" connected input emitter followers. This feature minimizes the "off-null" current, and prevents possible damage to sensitive devices under test.

For details, circle No. 66 on Reader Information Card.



CONVAIR PNEUMATIC SHOCK TESTER

Engineers at Convair (Astronautics) Division of General Dynamics Corp., San Diego 12, Calif., have designed a pneumatic shock tester which can be utilized to test a variety of electronic components, according to an announcement by the company. It is said to be able to determine peak shock in a part before failure.

A pneumatic pressure chamber in the shock tester can withstand 75 psi. A piston, mechanically restrained against the pressure until released, pushes against the bottom of a shock table holding the specimen.

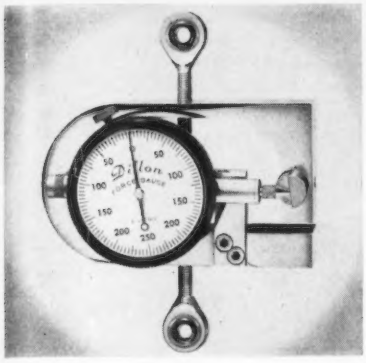
The shock caused by the sudden release of the piston generates the desired number of g's for the desired length of time. After release of the piston and the initial acceleration, the shock table coasts to a stop vertically.

For details, circle No. 67 on Reader Information Card.

PUSH-PULL GAGE

The availability of a new push-pull gage is announced by W. C. Dillon & Co., Inc., 14620 Keswick St., Van Nuys, Calif. According to manufacturer, it does the work of two separate compression and tensile measuring devices. The split dial gives readings of compression loads and tensile loads. Also, it can be hooked into a system to measure and control oscillating loads in compression and tensile.

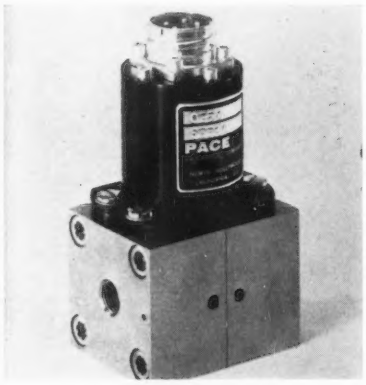
Applications for this unit are said



to be found wherever pressure, tensile or torque loads are to be measured. Testing shut-off valves, checking calibration of other devices, measuring tension in moving wire, testing mechanical efficiency of hydraulic door closers, preventing overloading of conveyor systems and setting spring adjustments are typical of its uses.

All units can be supplied with microswitches as optional features.

For details, circle No. 84 on Reader Information Card.



PRESSURE TRANSDUCER

Pace Engineering Co., 13035 Saticoy St., North Hollywood, Calif., announces the production of a new, d-c output variable reluctance, differential pressure transducer. Its ac–dc input circuit permits operation from either the 115 v line or 28 v dc. The manufacturer also states that Model CP53D is capable of extremely low differential pressure measurement in the presence of line pressures up to 5000 psi, with the accuracy required for laboratory and industrial measurement. Pressure ranges of the new instrument are from ±0.1 to ±2500 psig.

Accepting corrosive liquids and gases, both sides, Model CP53D transducers are said to withstand extreme mechanical shock and vibration.

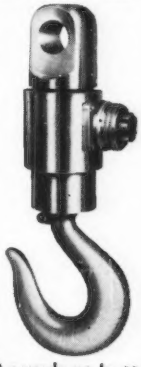
For details, circle No. 85 on Reader Information Card.

READS DIRECTLY IN POUNDS NO CIRCUIT BALANCING. NO CHARTS. CHOICE OF TENSILE COMPRESSION OR PUSH-PULL.

Designed upon differential transformer principle. Has no internal gears, levers or linkage to bind or wear.

Dillon Load Cells are accurate to within ½ of 1% of full capacity at any point on dial. Non-magnetic. Sensitive to load change of 1/10th of 1% of scale capacity. Tare loads up to 30% can be quickly canceled. Linear dial spacing made possible due to fact that output from Load Cell is in exact linear relation to mechanical displacement.

12 volt DC battery operated model or straight AC model optional. As many as four Load Cells may be used with one Indicator which totalizes individual loads. Minute elongation of Load Cell, even when operating at peak, varies between .002" and .004" depending upon capacity.



Load Cells are fully sealed. Most capacities are of beryllium copper. Can't rust or corrode. Explosion-proof versions can be supplied on order. AN type sockets and plugs used throughout with heavy duty cable.

Temperature compensated circuit utilizes semiconductors entirely. No tubes. No complicated, troublesome amplifiers. Load Cell safety factor is 4-1. Hysteresis is less than 1/10th of 1% of full scale capacity. Recorder jack is standard.

27 DIFFERENT CAPACITIES from as low as 0-150 pounds up to as high as 0-200,000 pounds. Variety of connectors such as swivel hooks, cable sockets, threaded studs, etc., available as accessories.

LC-2 COSTS YOU NOTHING TO INVESTIGATE. WRITE TODAY FOR PROFUSELY ILLUSTRATED BROCHURE.

W. C. Dillon & CO., INC.
WRITE DEPT. J 9
14576 KESWICK STREET
VAN NUYS 4, CALIF.
PHONE: STate 5-3186

For details, circle No. 12 on Reader Information Card

Multirecord Autographic Plotters

A new brochure, Bulletin PG-100, on their complete line of multirecord autographic plotters is being offered by Gilmore Industries, Inc., 13015 Woodland Ave., Cleveland 20, Ohio.

Bulletin PG-100 contains application data and complete specifications on Gilmore standard, high-speed, and extra-high-speed strain gage plotters; modular plotter; segmental recorder and scanner plotter.

Savings in test time and fast evaluation of tests obtained with "visual data plotting" of Gilmore plotters are also explained.

For your free copy, circle No. 101 on Reader Information Card.

Couplings and Connectors for Vacuum Systems

Central Scientific Co., 1700 Irving Park Rd., Chicago 13, Ill., report that their Booklet 327 covers a new and complete line of vacuum-system couplings and connectors.

The line includes couplings for standard tubing, feed-through electrodes, connectors, gage ports, release valves, caps and nipples and reducers—important components to construction of vacuum systems for research and process applications.

For your free copy, circle No. 102 on Reader Information Card.

Heavy-duty Mount

Bulletin 60-04.3, published by Barry Controls, Division of Barry Wright Corp., 700 Pleasant St., Watertown 72, Mass., contains complete information on the new Barry heavy-duty mount, series HM.

The manufacturer reports that this mount is designed to support loads up to 128,000 lb per mount, isolating the shock and vibration of mechanical and hydraulic presses of 200 ton capacity and greater. Each serialized HM mount is custom engineered for the particular machine and pit.

Bulletin includes photographs, dimension drawings, technical data on application and lists typical equipment that can be advantageously supported on this mount.

For your free copy, circle No. 103 on Reader Information Card.

Preferred Circuits Handbook

HANDBOOK PREFERRED CIRCUITS, Navy Aeronautical Electronic Equipment, NAVWEPS 16-1-519, "Supplement No. 3," 100 pages, $0.55, dated Apr. 1, 1960, issued September 1960. (Available from the Superintendent of Documents, U. S. Government Printing Office, Washington 25, D. C.) This latest supplement to HANDBOOK PREFERRED CIRCUITS contains seven germanium transistor circuits (PC 210 through 216), two silicon transistor circuits (PC 252 and 253), and one vacuum-tube instrument servo circuit (PC 79). Preferred circuits 210–216 are a compatible set of transistor digital logic circuits for use in computer, control and communication equipment. PC 252 is a variable gate generator, and PC 253 is a nonsaturating bistable multivibrator capable of operating at rates up to 1 mc. The vacuum-tube circuit, PC 78, is a phase sensitive null detector for use in servo control circuits. The looseleaf format of Supplement 3, like the original HANDBOOK and Supplements 1 and 2, allows the new circuits to be added to the handbook.

VHF Frequency Meter

Two-color data sheet describing a continuously variable frequency meter designed to measure and generate over a frequency range from 20 to 1000 mc, with guaranteed accuracy of 0.0001%, is available from Gertsch Products, Inc., 3211 S. La Cienega Blvd., Los Angeles 16, Calif. Also included in the data sheet, designated as Bulletin FM-7, are detailed technical specifications, and product illustrations.

According to manufacturer, Model FM-7 meets the FCC tolerance requirement of ±0.0005% with a minimum safety factor of 5 to 1. When combined with the DM-3 deviation meter and RFA-1 attenuator, unit provides a complete communication service package. Both portable and rack mounted versions are available.

For your free copy, circle No. 104 on Reader Information Card.

Automatic Chart Changer

Barton Instrument Corp., 580 Monterey Pass Road, Monterey Park, Calif., has introduced an automatic chart changer for circular chart recorders.

Intended for application in all fields where the measurement and recording of gases and fluids is accomplished through the circular chart type recorder, this chart changer makes it possible for a recorder to operate unattended for up to 16 days, using a new 24-hr chart for each day, the manufacturer reports.

A bulletin covering this chart changer is available.

For free copy, circle No. 105 on Reader Information Card.

Servomotors

A catalog showing new models and up-dated specs on Size 8 and Size 11 servomotors has just been published by Helipot Division of Beckman Instruments, Inc., 2500 Fullerton Rd., Fullerton, Calif.

The 24-page, 2-color catalog shows revised model numbers, complete electrical and mechanical specifications, outline drawings and torque-speed curves. Both the Size 8 and Size 11 lines include, as standard units servomotors, velocity-damp servomotors, inertia-damp servomotors and servomotor-generators with optional voltage requirements and mechanical characteristics.

Of special interest to systems engineers is a two-page technical discussion of damping theory.

For your free copy, circle No. 106 on Reader Information Card.

SOCIETY FOR EXPERIMENTAL STRESS ANALYSIS
21 Bridge Square, Westport, Connecticut

SESA PUBLICATIONS

PROCEEDINGS OF THE SOCIETY FOR EXPERIMENTAL STRESS ANALYSIS

The PROCEEDINGS are a series of bound volumes comprised of the technical papers presented at National Meetings. These contain the results and descriptions of analytical and experimental investigations, as well as techniques for the measurement of mechanical strain. Published twice annually, they include up-to-date information on various phases of instrumentation. Currently available:

Volume III, No. 2
Volume IV, No. 1
Volume IV, No. 2
Volume V, No. 1
Volume XI, No. 2
Volume XII, No. 1
Volume XII, No. 2
Volume XIII, No. 1 (limited)
Volume XIII, No. 2 (limited)
Volume XV, No. 2
Volume XVI, No. 2
Volume XVII, No. 1
Volume XVII, No. 2

CUMULATIVE INDEX—PROCEEDINGS

Thirty-four books of the PROCEEDINGS have been published to date. This Index has been prepared to serve as an easy reference to any one of the 546 papers which have been published in the first thirty books. It consists of three parts: (1) chronological index, (2) author index, (3) topical index. It represents a "master reference file" to all PROCEEDINGS published, from Vol. I, No. 1 through Vol. XV, No. 2.

HANDBOOK OF EXPERIMENTAL STRESS ANALYSIS

This book opens with a thorough presentation of mechanical properties. It presents practical, usable information on: mechanical, electrical and optical gages, brittle coatings, photoelasticity, X-rays, service fractures, analogies, vibration measurements, theory of elasticity and precision of measurements. Stiff cloth binding—contains 1060 pages of text.

MANUAL ON EXPERIMENTAL STRESS ANALYSIS TECHNIQUES

For those who are new to the field of experimental stress analysis, the MANUAL introduces the fundamentals of experimental stress analysis in terms easily understood by the novice. It contains detailed instructions and, with the aid of illustrations and charts, elaborates on the application of three particular methods for determining stress: (1) SR-4 bonded wire strain gages, (2) Stress-coat (brittle lacquer) and (3) photoelasticity.

STRAIN GAGE TECHNIQUES

Twenty-three chapters of lectures and laboratory experiments of special summer programs presented at leading universities, covering such topics as: fundamental concepts, selection of gages, basic considerations in strain-gage instrumentation, electrical and mechanical aspects of the gage system, the potentiometric circuit, the wheatstone bridge, amplitude modulation, fundamentals of measuring systems, some commercial systems and rosette analysis. Approximately 680 pages.

CATALOG OF EQUIPMENT USED IN EXPERIMENTAL STRESS ANALYSIS

The Catalog contains references to manufacturers of equipment, hardware and instrumentation used in the measurement of strain.

EXPERIMENTAL MECHANICS

Official monthly journal of the SESA. Includes technical papers and news items of interest to SESA members and to industry.

ORDER FORM

Publication	Author/s or Editor/s	Members' Price	Bulk Orders*	Price per Copy (Non members)	Postage and Handling (Add for Each Copy) U. S.	Foreign	State Volume Number	Specify Quantity	Total
Proceedings of the Society for Experimental Stress Analysis	Mahlmann, CV and Murray, WM	$ 6.00	$ 6.00	$ 7.00	25¢	50¢	_____	_____	$_____
Handbook of Experimental Stress Analysis	Hetenyi, M.	16.75	16.75	17.75	Included	85¢		_____	$_____
Manual on Experimental Stress Analysis Techniques	Perry, CC and Bean, WT	2.00	1.25	3.00	Included	10¢		_____	$_____
Strain Gage Techniques	Murray, WM and Stein, PK	12.00	12.00	13.00	Included	85¢		_____	$_____
Catalog of Equipment	Hofmeister, WF	2.00	1.25	3.00	Included	10¢		_____	$_____
Cumulative Index		2.00	1.25	3.00	Included	10¢		_____	$_____
EXPERIMENTAL MECHANICS**	Monthly Journal	1.00		1.50	Included	20¢		_____	$_____

*Bulk Orders—10 or more copies of a *single book* (for *PROCEEDINGS*—10 or more copies of a single volume and number).
**Annual Subscription (including postage)—Domestic, $10; Foreign, $12.

Send books to:

_____
_____
_____
_____

Send bill to:

_____
_____
_____
_____

Total.....$_____
Remittance Enclosed ☐ Bill me ☐
PAYMENT MUST ACCOMPANY ORDERS FOR LESS THAN $5.00
Order Number_____
Order Issued by_____
Date_____

CHAPMAN LABORATORIES

PHOTOELASTIC POLARISCOPES

MODEL MAKING KITS

BOX 207
WEST CHESTER
PENNSYLVANIA

For details, circle No. 6 on Reader Information Card

Experimental Study of Large-diameter Thin-wall Pressure Vessels

Brittle coatings and electrical strain gages are employed to determine stress distribution over surfaces of vessels

by A. J. Durelli, J. W. Dally and S. Morse

ABSTRACT—An experimental stress analysis of three cylindrical pressure vessels with radius/thickness ratios ranging from 100 to 238 and different head closures is described. Brittle coatings and electrical strain gages were employed to determine stress distributions over the entire outer surface of the vessels. Electrical strain gages alone were used to determine stresses on the inside surface of the vessels. Particular emphasis was placed on determining stress concentrations and on nonlinear effects produced by geometric imperfections. An attempt was also made to correlate the failure, which started in the cylindrical portion of the three vessels, with the elastic-stress distribution. It was found that the imperfections in the cylinder were not significant if the vessel was fabricated from a ductile steel. However, if the vessel was constructed from a high-strength but brittle steel, the imperfections significantly lower the bursting strength of the vessel.

Introduction

The authors believe that a recent investigation they have conducted dealing with large-diameter thin-wall vessels may be of interest to people working in the missile field. Three different vessels were studied experimentally in order to determine their stresses and failure characteristics when subjected to internal pressure. The results of the investigation were used as a basis for developing lightweight vessel designs and for evaluating various manufacturing techniques used in their fabrication.

Although the experimental procedure varied somewhat in the investigation of each of the three vessels, the following three phases were common to each analysis. First, brittle coatings were applied to determine the stress distributions over the entire outer surfaces. Second, certain regions of interest indicated by the brittle-coating analysis were instrumented with electrical-resistance strain gages and the response of these gages was followed within the elastic range of the material. Third, the vessels were pressurized until failure occurred and the load, the location of the failure and the type of the failure were recorded.

In this presentation, emphasis will be placed on the experimental procedures and general findings rather than specific results which hold only for a particular vessel.

Vessel A

Vessel Description

Vessel A was about 7 ft long with a diameter of about 28 in. and a wall thickness of about 0.110 in, giving an R/t ratio of 128 in the cylindrical portion of the vessel. The forward end of the vessel was hemispherical with a wall thickness of about 0.15 in. and a circular opening at the top. The aft end of the vessel was in the form of a knuckle ring of variable thickness with a large circular opening. In the experimental analysis, both of the openings at the ends of the vessel were closed with flat plates and sealed with O rings. The vessel was fabricated by drawing two ends as cups and welding these two ends together with a circumferential weld. The vessel material was AISI 4130 steel heat treated to a hardness of Rockwell C-37. Tests of a coupon taken longitudinally from the vessel after its failure gave a 0.2% yield strength of 170,000 psi and an ultimate strength of 190,000 psi. An over-all view of the vessel is given in Fig. 7. Two thin skirts welded around the knuckles hide partially the forward and aft closures. These skirts are stress free.

Outside micrometer measurements at a number of positions about and along the axis of the cylinder indicated that the vessel was not perfectly cylindrical. Two types of imperfections occurred. The first type may be defined as "out of roundness," indicating that the vessel is somewhat elliptical in shape rather than circular. The second type may be called "corrugation," indicating local irregularities

A. J. Durelli and S. Morse are associated with Armour Research Foundation, Chicago, Ill. Dr. Durelli is also Professor of Civil Engineering at the Illinois Institute of Technology. J. W. Dally, formerly with Armour Research Foundation, is now Assistant Professor in Mechanics at Cornell University, Ithaca, N. Y.

Paper presented at 1959 SESA Spring Meeting held in Washington, D. C., on May 20–22.

in the cylindrical shell. It will be shown later that these imperfections produce stress concentrations at various portions of the cylindrical shell.

Measurements of the wall thickness at about 200 points were made employing ultrasonic methods. These measurements indicated that the thickness of the drawn shell was not constant, but varied from 0.099 to 0.123 in., a difference of more than ±10% from the specified thickness. Thickness measurements were recorded on the surface of the vessel, as on position 4 in Fig. 4.

Brittle-coating Analysis

The entire outer surface of Vessel A was coated with Stresscoat 1209 to a thickness of about 0.008 in. The coating was cured at 86° F for 14 hr and cooled slowly to a test temperature of 74° F over an 8-hr interval. The test was conducted using conventional techniques as described elsewhere.[1] Isostatics and isoentatics were obtained over the entire outer surface, as is illustrated in Fig. 1, which shows the brittle-coating crack formation on the forward portion of the cylindrical shell. An examination of the isoentatic lines shown in this figure indicates that an appreciable nonuniformity in the distribution of stresses occurs over the cylindrical portion of the vessel. If the stresses in the vessel occur as the elementary shell theory indicates, the brittle coating should crack over the entire cylindrical portion of the vessel at the same load. Actually, the coating first cracked at a pressure somewhat less than 125 psi and continued to crack over other areas until a pressure of 600 psi had been reached. The difference in the level of stress from one point to another, which in this case was as much as 5 to 1, was the result of imperfections in the geometry of the cylinder. A close-up view of the brittle-coating pattern in the neighborhood of two of these local imperfections is shown in Fig. 2.

As the pressure is increased, the actual geometry of the vessel approaches the ideal cylindrical shape. The local imperfections become less pronounced and have less of an effect on the stress distribution. Therefore, the stresses do not increase linearly with pressure. The nonlinear behavior of the stresses seriously impaired the usefulness of the brittle-coating method for providing quantitative data of

Fig. 1—Brittle-coating crack formation produced on the forward portion of the cylindrical shell by internal hydrostatic pressure. The lack of uniformity of the stress level is indicated by the brittle-coating isoentatic lines (Vessel A). (The top portion of the vessel, above the ring, is a stress-free skirt)

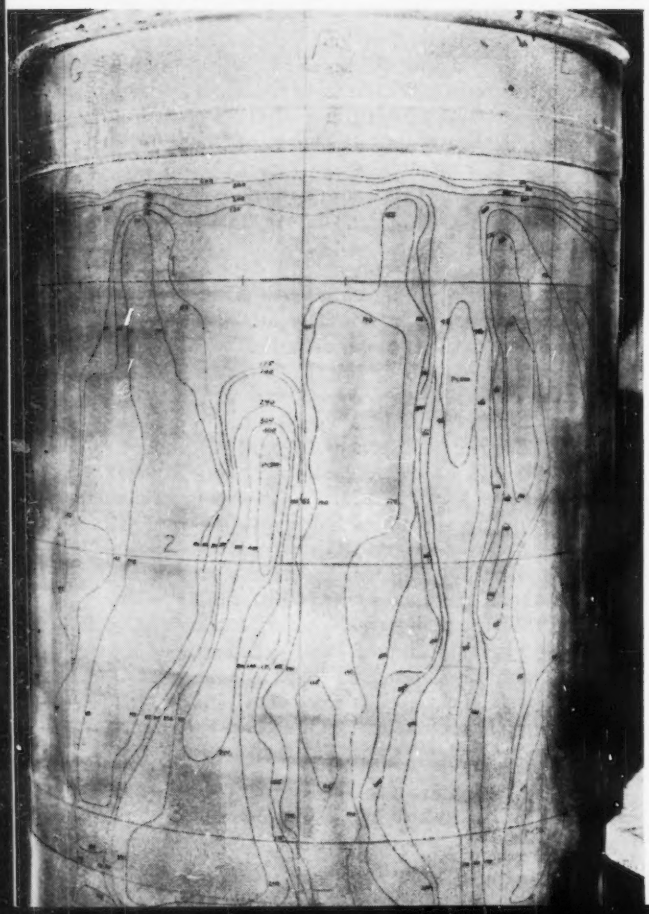
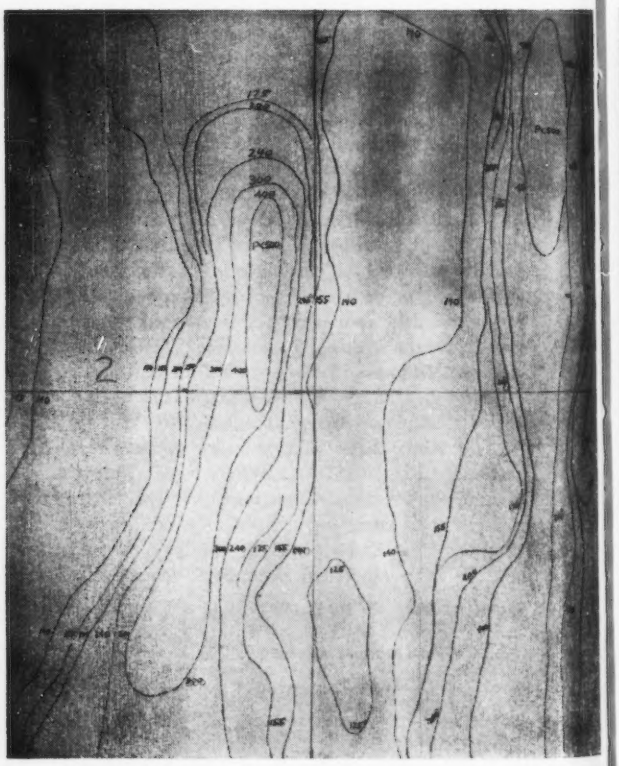


Fig. 2—Close-up view of the brittle-coating crack formation in an area where local cylinder imperfections influence the stress distribution (Vessel A). (The figures on the isoentatic lines correspond to the amount of pressure in psi needed to bring the cracks to that line)



the stresses at pressure levels above or below the values at which cracking of the coating occurred. In other words, the coating provides only one point on the curve showing stresses versus pressure at a given location.

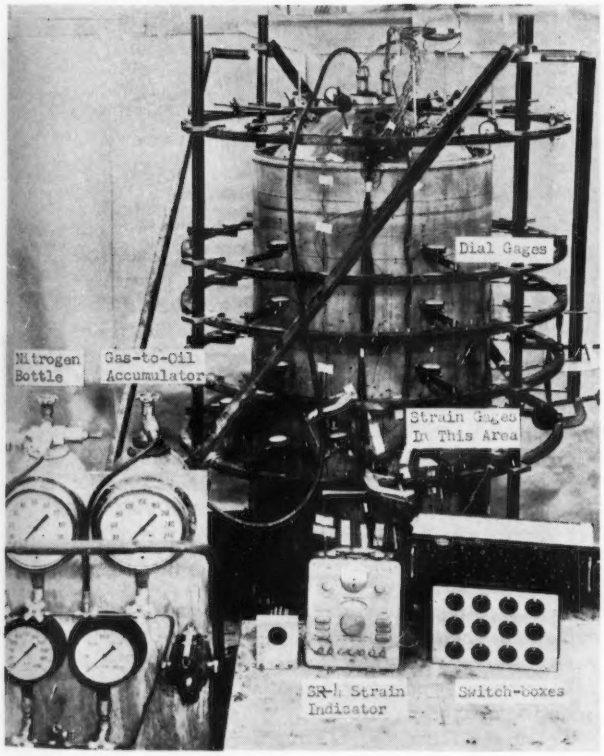
Strain-gage Analysis

Although the brittle-coating test provided much information concerning location, magnitude and directions of stresses, electrical strain gages were necessary to supplement the brittle-coating data. A few of these gages were placed at the zone of local imperfections and on the forward and aft ends where the coating response was not sufficient. In all a total of 72 BLH type A-7 gages were employed in the investigation.

Strain readings were taken at pressures of 50, 100, 150, 200, 300, 400, 500, 600 and 700 psi. The test was discontinued at a pressure of 700 psi because of a seal failure. The hydraulic-loading system, the strain-gage installation and instrumentation, as well as a dial-gage installation, are shown in Fig. 3.

The strain gages mounted on the internal and the external surfaces of the heads of the vessel indicated that both the bending and membrane stresses in the heads were relatively low in comparison to the cylinder stresses.

Fig. 3—Hydraulic-loading system, electrical-strain-gage installation and dial gages (Vessel A)



Strain gages were mounted on both inside and outside surfaces along a line drawn through two imperfections as illustrated in Fig. 4. By combining the data obtained from the inside and outside surfaces, it was possible to plot bending and membrane stresses in the hoop and axial directions as a function of pressure, as shown in Fig. 5. At position 2 for a pressure of 700 psi, the hoop stresses are in-

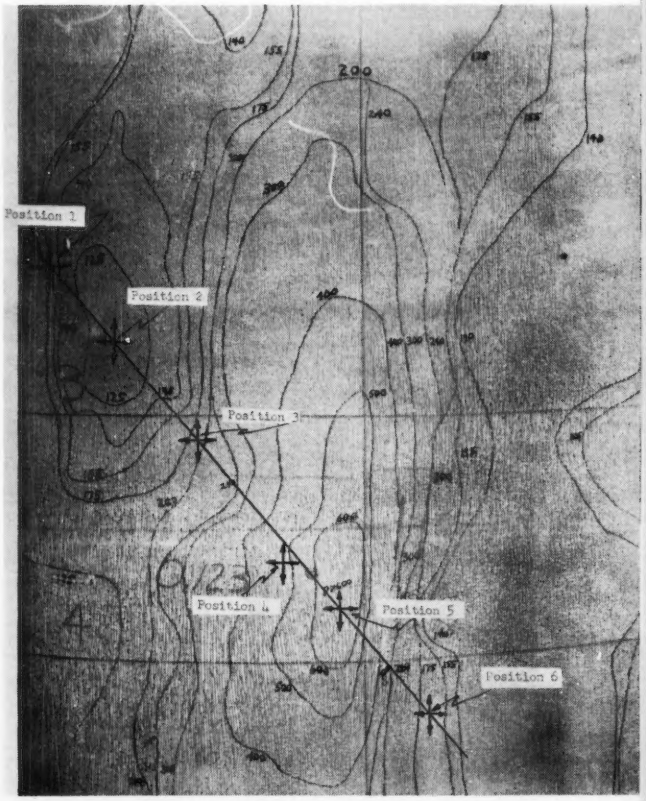


Fig. 4—Electrical-strain-gage positions in a region on the cylindrical shell where two local imperfections occur (Vessel A). (The figure next to Position 4 corresponds to the thickness of the vessel in inches)

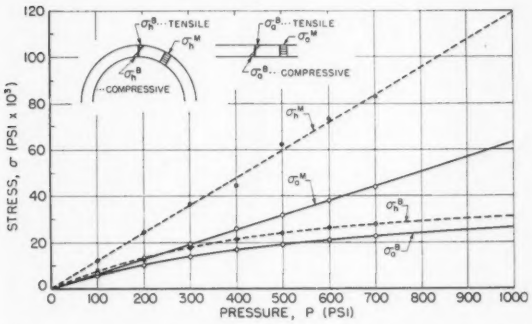


Fig. 5—Bending and membrane stresses as a function of pressure at Position 2 on the cylinder (Vessel A)

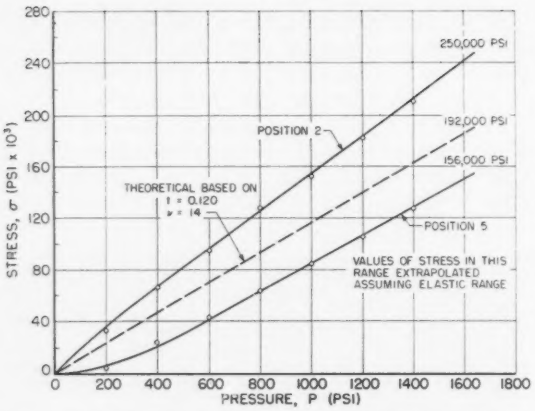


Fig. 6—Hoop stresses on the outside surface of Vessel A as a function of pressure at positions 2 and 5

Fig. 7—Over-all view of Vessel A showing the region of failure

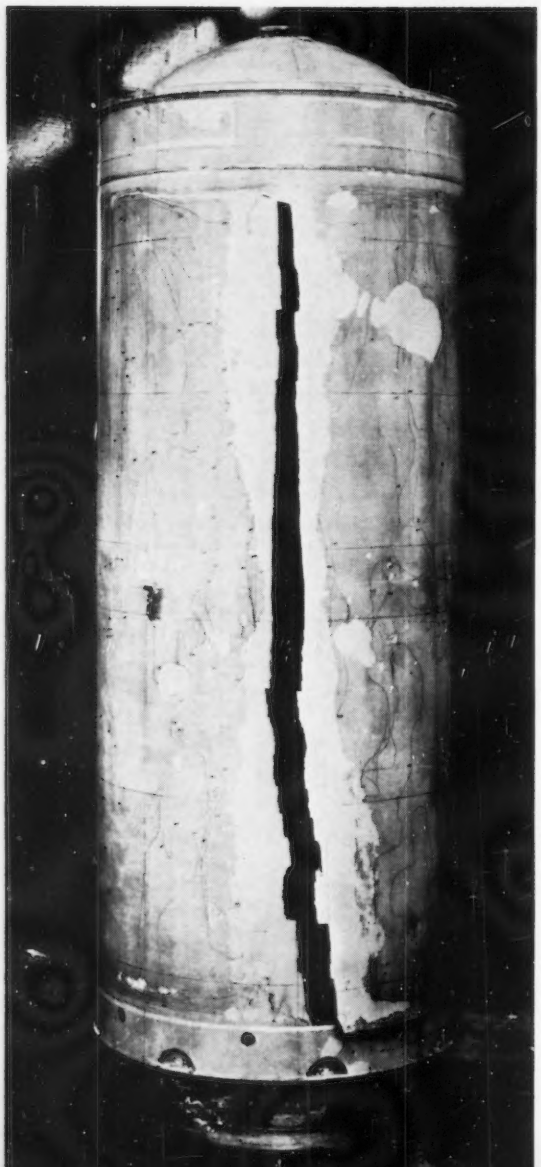


Fig. 8—Close-up view of the failure of Vessel A showing the fracture surfaces. (It is noted that the fracture surface is inclined 45 degrees to the surface of the vessel indicating a shear-type failure)

creased 34% from 82,000 psi (membrane) to 110,000 psi (total) by addition of the bending stresses.

Combined Strain-gage and Failure Analysis

The strain-gage analysis described in the previous section was extended in this phase of the investigation in order to study the changes in the stress distribution on the cylindrical shell when the internal pressure is increased up to 1600 psi. The strain-gage data were converted to stress data using Hooke's law. Results for the stresses at positions 2 and 5 are shown in Fig. 6, where hoop stresses have been plotted as a function of pressure and compared with the theoretical values. An appreciable difference exists between the stresses at positions 2 and 5. The stresses at position 2 for a pressure of 1600 psi are approximately 30% higher than the theoretical stresses; whereas, the stresses at position 5 are approximately 20% lower. These differences are due to bending stresses produced by imperfections in the cylindrical shell which the elementary theory does not consider. A change in curvature in the vessel will produce bending stresses which superimpose upon the membrane stresses.

The vessel failed by splitting longitudinally at a

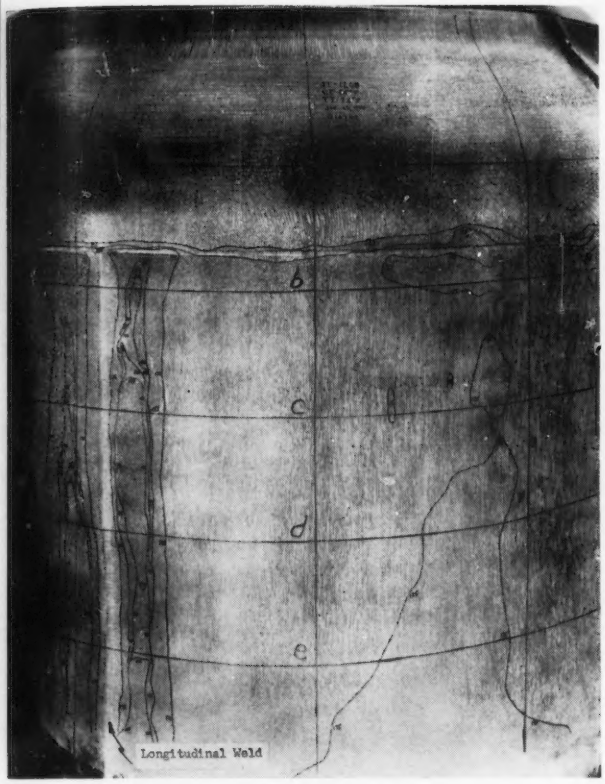


Fig. 9—Brittle-coating crack pattern produced over a large region of the vessel by internal hydrostatic pressure (Vessel B). (The figures associated with each isoentatic represent the pressure in psi necessary to crack the coating at the points on the isoentatic. The isoentatic lines in the neighborhood of the longitudinal weld indicate a concentration of stress in the areas adjacent to the weld)

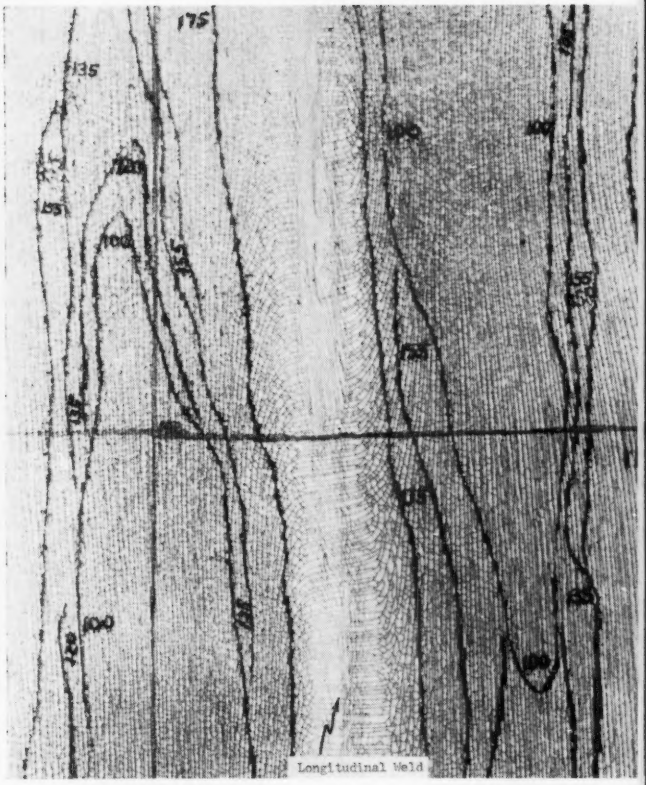


Fig. 10—Detail view of the brittle-coating pattern in the regions adjacent to the longitudinal weld (Vessel B). (The figures associated with each isoentatic represent the pressure necessary to crack the coating at the points on the isoentatic. Cracks on the weld, between the two 175 isoentatics were obtained by refrigeration. They show several changes in the direction of the maximum stress)

pressure of 1640 psi. An over-all view of the vessel showing the region of failure is presented in Fig. 7, and a close-up view showing detail of the fractured surface is shown in Fig. 8. The fracture surfaces are inclined at a 45-deg angle with the outer normal to the vessel surface. This angle of inclination indicates that the type of failure was shear. It is believed that the failure initiated near the circumferential weld and the crack propagated in both the forward and aft directions approximately parallel to the axis of the vessel. When the cracks reached the head closures, they turned abruptly through 90 deg and began to propagate circumferentially about the forward and aft extremities of the cylindrical shell.

Correlation Between Elastic Stress Analysis and Failure

The Hencky-Von Mises theory of failure[2], which is based on strain-energy principles, is considered as one of the most applicable for predicting ductile failure. The expression relating the stress at failure σ_u and the principal stresses σ_1, σ_2 and σ_3 is given below:

$$\sigma_u = \frac{1}{\sqrt{2}} \sqrt{(\sigma_1 - \sigma_2)^2 + (\sigma_1 - \sigma_3)^2 + (\sigma_2 - \sigma_3)^2} \quad (1)$$

If it is assumed that the shell is perfectly circular and constant in thickness, the principal stresses are:

$$\left.\begin{aligned} \sigma_1 &= \sigma_h = \frac{pr}{t} \\ \sigma_2 &= \sigma_a = \frac{pr}{2t} \\ \sigma_3 &\approx 0 \end{aligned}\right\} \quad (2)$$

where p is the vessel pressure, psi, r is the vessel radius, in. and t is the vessel thickness, in. By substituting eq (2) into eq (1) it follows that:

$$\sigma_u = \frac{\sqrt{3}}{2} \frac{pr}{t} \quad (3)$$

or the bursting pressure is given by

$$p = \frac{\sigma_u t}{r} \frac{2}{\sqrt{3}} \quad (4)$$

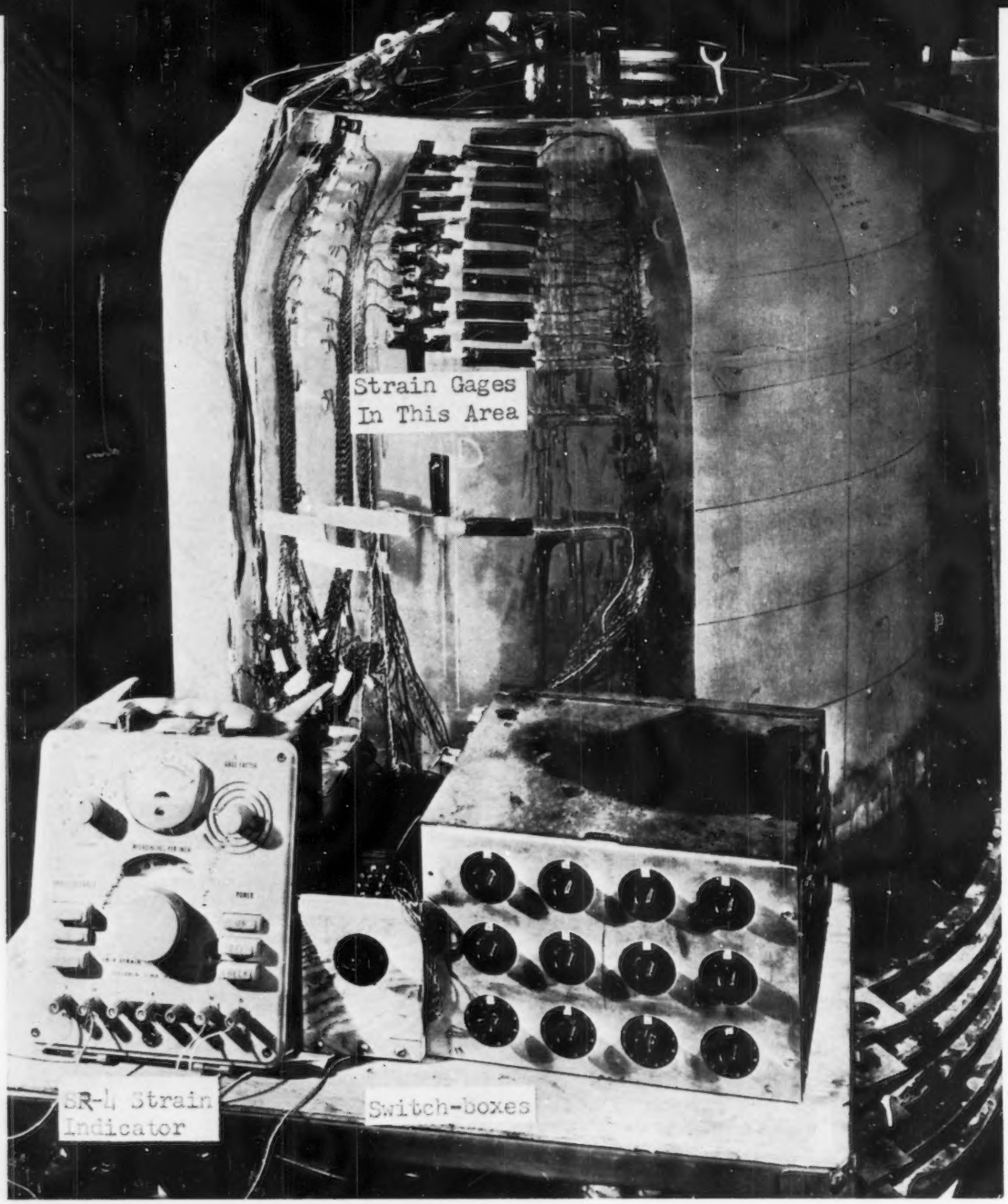


Fig. 11—Strain-gage installation and recording equipment employed with Vessel B

The wall thickness and the radius near the failure location were about 0.100 and 14 in., respectively. Hence, from eq (4), the bursting pressure was computed as 1570 psi. This value is about 4% lower than the 1640 psi actually required to fail the vessel.

This analysis indicates that the local imperfections which influence the elastic stresses very appreciably do not significantly influence the bursting pressure. The vessel material was sufficiently ductile to yield before failure and redistribute the bending stresses. However, this redistribution of the bending stresses is dependent on material ductility, and, if very high-strength steels with low ductility are employed in fabrication, the local cylinder imperfections may become significant.

Vessel B

Vessel Description

Vessel B was about 40 in. long, with a diameter of 38 in. and a wall thickness of 0.080 in., giving an R/t ratio of 238. The cylindrical shell was formed by rolling two flat sheets of steel and welding them along the longitudinal axis. The forward end was ellipti-

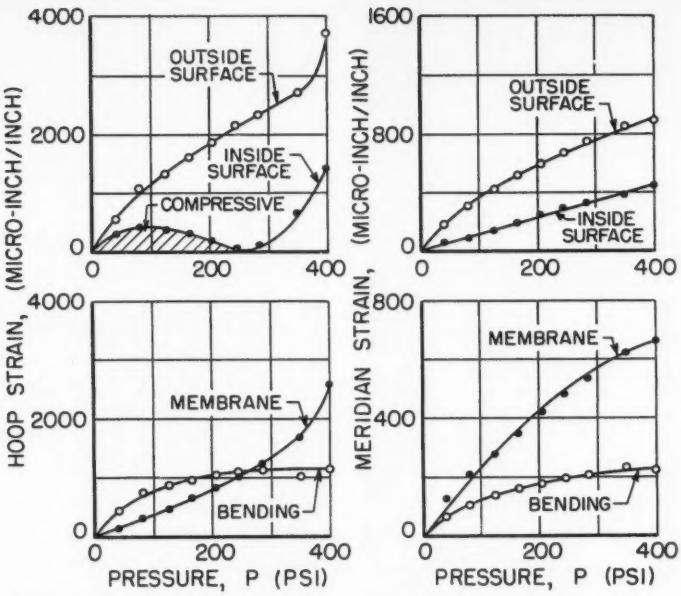


Fig. 12—Strains as a function of pressure at the region of stress concentration to the right of the longitudinal weld in Vessel B

cal in shape and deliberately overdesigned since it was not the subject of the investigation. The aft-end knuckle was machined from a forged ring to specified dimensions and joined by a girth weld to the cylindrical portion of the vessel. The opening was sealed with a flat plate and O ring assembly (Fig. 11).

The vessel was fabricated from AISI 4130 steel, stress relieved after welding and not hardened. Material tests indicated a 0.2% yield strength of 103,000 psi and ultimate strength of 123,000 psi. Hardness tests showed the stress-relieved steel had a Rockwell hardness of C-22.

Micrometer measurements again indicated that the cylindrical shell was not perfectly cylindrical. No thickness measurements were made. It is believed that the sheets of rolled steel used in the vessel fabrication would be nearly constant in thickness.

Brittle-coating Analysis

Stresscoat ST-1208 was sprayed over most of the surface of Vessel B. The coating was cured at 80° F for 14 hr, and then cooled slowly to a testing temperature o 72° F for 7 hr.

The vessel was pressurized in incremental steps until coating cracks had covered the entire cylindrical body and a portion of the aft-head knuckle. Since it was not safe to further increase the pressure, coating response was obtained in the other regions by refrigerating the coating

Typical brittle-coating results are illustrated in Figs. 9 and 10, where photographs of the crack formations are shown. In Fig. 9, three items of importance are noted. First, the regions on either side of the longitudinal weld show a very high concentration of stress. Second, the aft knuckle is stressed to a much lower level than the cylindrical portion of the vessel. Last, two isotropic lines are clearly indicated in the aft-knuckle region where the maximum principal stresses change direction twice.

A detail view of the brittle-coating crack pattern in regions adjacent to the longitudinal weld is shown in Fig. 10. The complex state of stress along the weld is clearly indicated by this crack pattern. The bending stresses superimpose upon the membrane stresses to produce high tensile stresses on the outside surface of the vessel at the regions to the left and right of the weld and to produce high tensile stresses on the inside surface at the weld itself.

Strain-gage Analysis

In order to supplement the brittle-coating data in the low-stressed regions of the aft knuckle and the highly stressed regions near the weld, a number of strain gages were mounted on both the inside and outside surfaces of the vessel. The vessel was pressurized to 400 psi and strain-gage readings were taken at a number of levels of pressure. The test setup showing the strain-gage installation and vessel configuration is shown in Fig. 11.

The data obtained in the knuckle region indicated a precise level of the stresses and gave a basis for making space-saving recommendations regarding the redesign of the aft knuckle.

Typical results from gages placed at one point on the right side of the weld where there is a concentration of stresses are shown in Fig. 12. An examination of these figures shows the very appreciable influence of the bending stresses. For example, the hoop bending strains were equal to the membrane strains at about 270 psi. The nonlinear behavior of the bending strains is due to changes in the radius of curvature of the local imperfections as the vessel is pressurized. The nonlinear behavior of the membrane stresses is presumably due to the yielding of the vessel which occurs at about 300 psi.

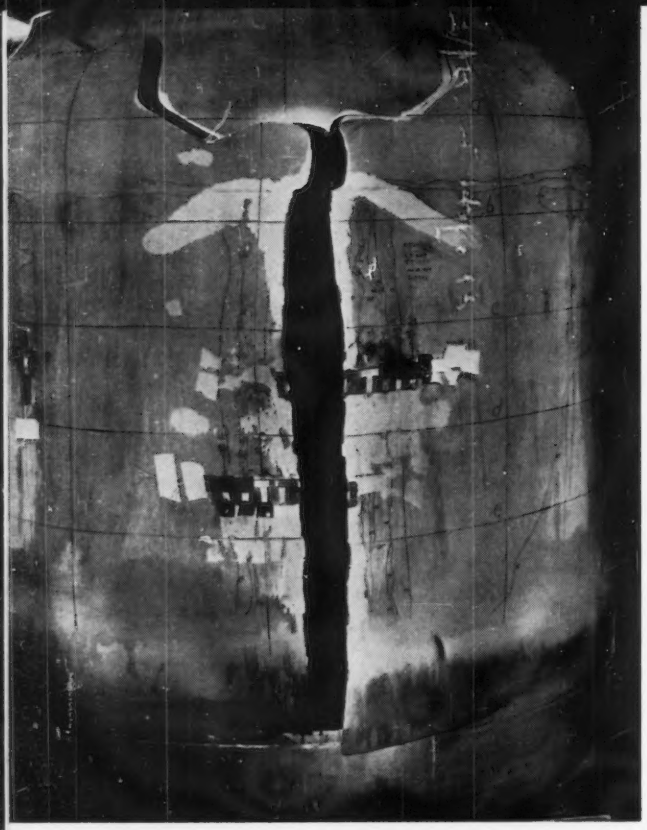
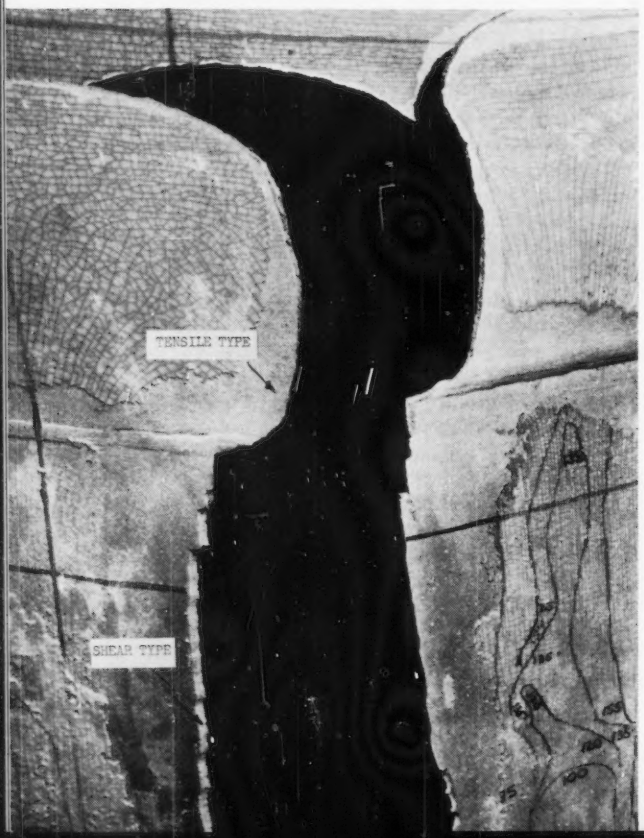


Fig. 13—Over-all view of the failure of Vessel B

Fig. 14—Transition from shear to a tensile-type failure as the crack propagated from the circular cylinder into the aft-end knuckle (Vessel B)



Failure Analysis

Vessel B was pressurized to 540 psi before it split along one of its longitudinal welds. An over-all view of the failure is shown in Fig. 13 and a detail view is shown in Fig. 14. An inspection of the fracture (see Fig. 13) showed that failure originated at about the center of the vessel and the crack propagated both forward and aft along the weld seam. When the cracks reached the head closure, they divided, turned and began to propagate circumferentially about the forward and aft extremities of the vessel. The fracture surfaces shown in Fig. 14 indicate a shear-type failure in the cylinder and a tensile-type failure in the low-stressed aft-knuckle region.

The ultimate strength, wall thickness and cylinder radius were substituted into eq 4, as in the case of Vessel A, and a bursting pressure of 600 psi was obtained. This value of 600 psi was about 10% higher than the 540 psi at which the vessel actually ruptured. This decrease in strength was probably due to the stress concentrations at the weld.

Vessel C

Vessel Description

Vessel C was about 3 ft high, with a diameter of 12 in. and a wall thickness of 0.060 in., which gives an R/t ratio of 100. The cylindrical portion of the vessel was made by roll forming a sheet of Vascojet-1000 steel and by welding it longitudinally. Hemispherical heads were welded to the cylindrical shell by means of girth welds (Fig. 15).

The vessel was subjected to the following heat treatment: (*a*) soaked at 1700° F for 1 hr, (*b*) transferred to a cyclone furnace at 800° F and held there until equalized, (*c*) air cooled at room temperature and (*d*) double tempered at 1000° F. The material exhibited a 2% yield strength of 198,000 psi and an ultimate strength of 237,000 psi with a Rockwell C hardness of 49 to 51.

Brittle-coating and Strain-gage Analysis

Brittle-coating and strain-gage tests were conducted with Vessel C in the same manner as previously described in the sections on vessels A and B. Typical brittle-coating results are shown in Figs. 15 and 16. More information could be obtained from the brittle-coating analysis conducted on Vessel C than from the analysis conducted on the other two vessels. The length of this paper prohibits, however, more than a few remarks concerning this analysis. The over-all crack patterns indicated in Fig. 15 show high stress concentrations in the region near the longitudinal weld. The detailed crack pattern shown in Fig. 16 indicates the completeness of the whole-field character of the brittle-coating approach. The primary isoentatics, shown by the solid scribed lines on the coating, give the maximum principal stresses, for a given pressure, and the secondary isoentatics shown by the dashed scribed lines give the minimum principal stresses. The direction of these

stresses is given by the cracks (isostatics). The use of the secondary isoentatic is not common among experimentalists. No cracks appear on the surface of the weld indicating that very low tensile stresses or likely compressive stresses are present on the outside surface of the vessel at these points.

Strain gages placed at the stress concentrations near the longitudinal weld again indicated that the local irregularities in the cylindrical shell produce appreciable increases in elastic stresses. This fact is substantiated by the results of a pair of gages placed at the concentration to the left of the longitudinal weld, as shown in Fig. 17, where the actual stress recorded was nearly twice the theoretical value.

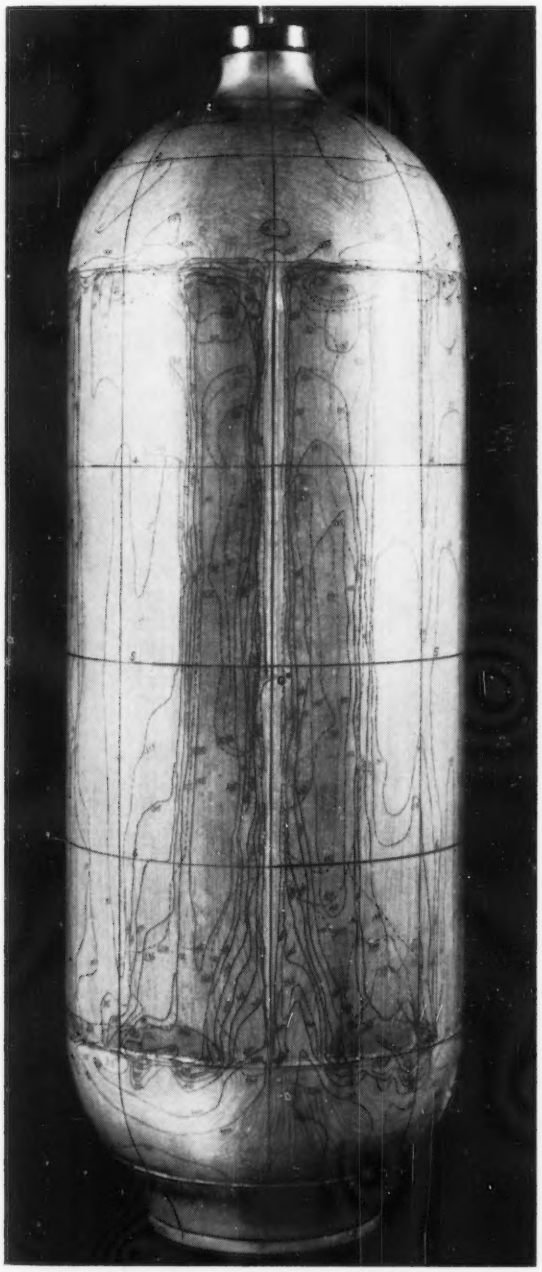


Fig. 15—Over-all view of the brittle-coating crack pattern on Vessel C

Failure Analysis

Vessel C failed violently at a pressure of 1500 psi. A photograph (see Fig. 18) taken by microflash lighting a few milliseconds after failure shows some fragmentation and the violence of the failure. The pieces of the vessel were recovered and the vessel was reconstructed as shown in Fig. 19. The herringbone appearance of the cracks indicates failure occurred along the weld at the position shown in Fig.

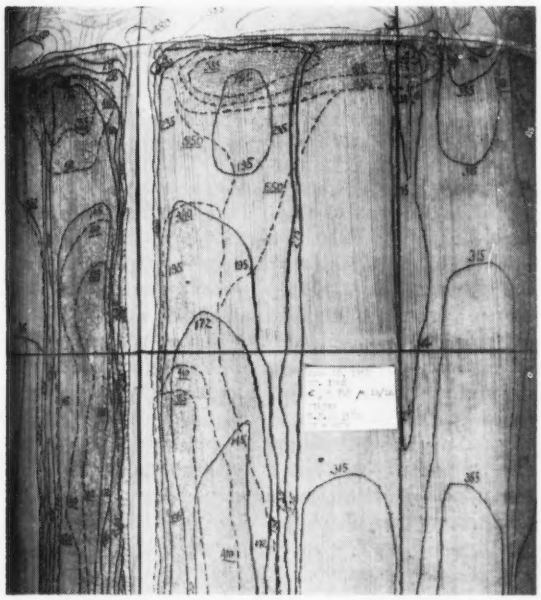


Fig. 16—Brittle-coating crack pattern over the forward section of the cylindrical shell near the longitudinal weld (Vessel C). (Solid isoentatics give maximum principal stresses. Dotted isoentatics give minimum principal stresses)

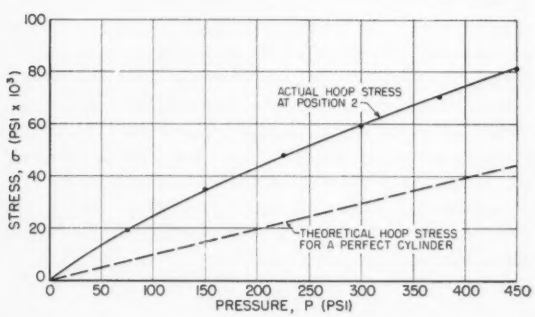


Fig. 17—Comparison of the actual hoop stresses with the theoretical hoop stresses for a perfect cylinder representing Vessel C.

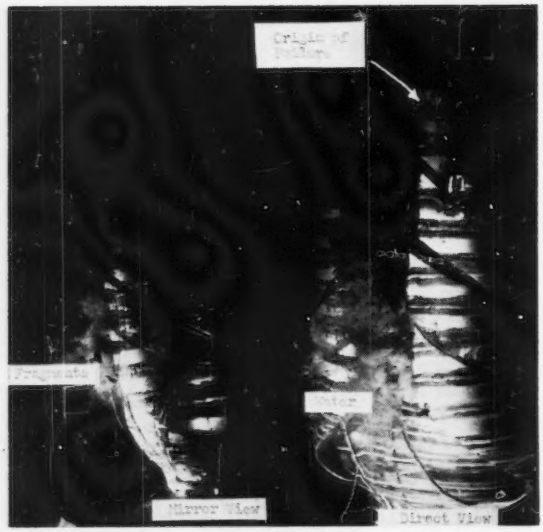


Fig. 18—Microflash photograph of Vessel C as it burst. The circular white lines are a spiral of silver paint, the breaking of which triggered the microflash

19. Examination of the fracture surfaces indicated the failure was a brittle- ensile type.

Equation 4 was again employed and gave a bursting pressure of 2790 psi which was about 86% higher than the 1500 psi at which the vessel actually failed. The premature failure was attributed to one or a combination of the two following causes: first, the stress level at the weld is much higher than indicated by the simplified theory, as evidenced in Fig. 17, and there was little or no redistribution of this concentration. Second, the weld efficiency was likely less than 100% and a reduction in strength resulted.

Conclusions

It was found in the stress analysis of three cylindrical pressure vessels with R/t ratios ranging from 100 to 238 that small local geometric imperfections influenced the elastic stress distribution very significantly. An attempt was made to correlate the failure phenomena with the elastic stress distributions. It was found that the local stress concentrations produced by these imperfections were significant in the failure of the vessel fabricated from a very high-strength brittle steel. For vessels constructed from lower-strength, but more ductile, steels the local concentrations of stress were not significant and bursting pressures could be predicted with fair accuracy using the Hencky-Von Mises theory of failure using the theoretical values of hoop and axial stress.

Acknowledgment

Credit should be given to Tahei Niiro, assistant experimental engineer, for spraying the brittle coating on the three vessels.

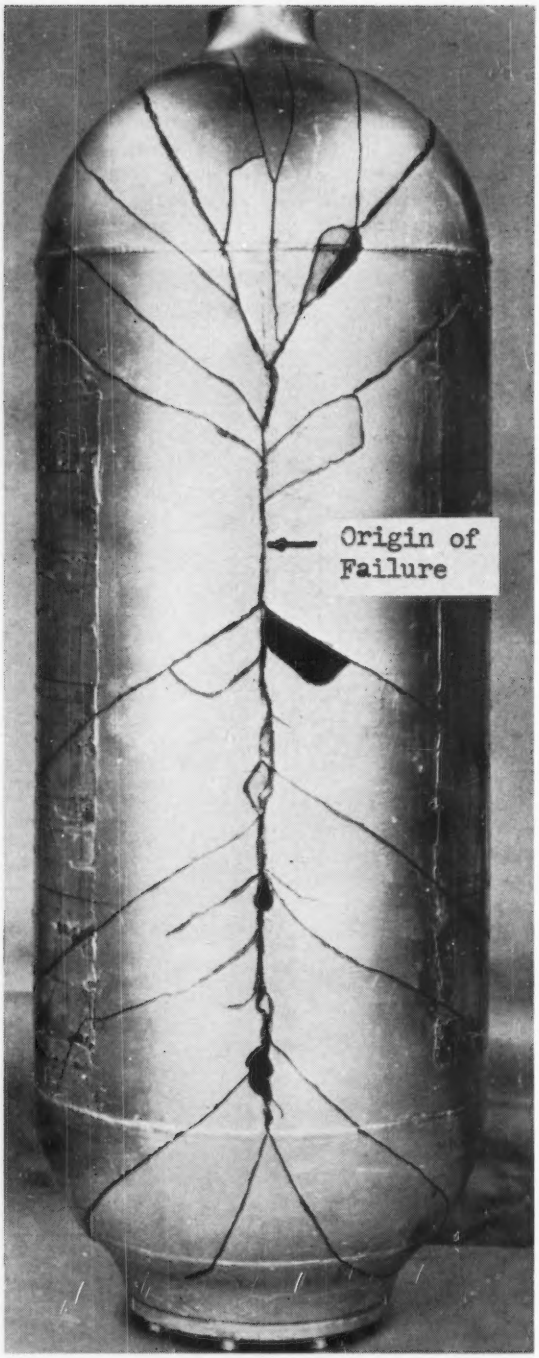


Fig. 19—Appearance of cracks produced by bursting Vessel C. (Reconstruction after failure)

References

1. *Durelli, A. J., Phillips, E., and Tsao, C., "Introduction to the Theoretical and Experimental Analysis of Stress and Strain," McGraw-Hill, 1958, Chap. 17.*
2. *Timoshenko, S., "Strength of Materials, Part II," 2nd Ed., p. 478.*

Experiments in Mechanical and Optical Coincidence in Photoplasticity

Experiments are aimed principally at answering the basic question of whether optical anisotropy is necessarily associated with mechanical anisotropy, a point of fundamental importance in the application of the shear-difference method

by M. M. Frocht and R. A. Thomson

ABSTRACT—In the extension of the shear-difference method to the separation of interior principal stresses in the elasto-plastic state,[1] a basic question arises whether the optical isoclinics give the directions of the principal stresses, i.e., whether optical coincidence exists. Experiments are described aiming to answer this question, and preliminary results are given for cellulose nitrate as a model material. Experiments are also described dealing with mechanical coincidence.

Introduction

This paper is a continuation of the studies in photoplasticity which the authors have been conducting for some time. In a previous paper[1] the authors applied the shear-difference method to the determination of stress distributions in bars with holes, grooves and fillets subjected to pure tension and stressed to the elasto-plastic state, and obtained substantial agreement with theoretical and experimental results. The use of the shear-difference method rests on the assumption that the isoclinic parameters give the directions of the principal stresses p and q.

However, it is known that, at least in materials like aluminum, mechanical coincidence, i.e., coincidence between the directions of p and q and the corresponding directions of the principal strains ϵ_p and ϵ_q, breaks down in all cases except when the directions of p and q are constant. No coincidence exists between the directions of the final stresses and strains, or between the directions of successive increments of stress and strain, when rotation of the stresses is present. In other words, after a first plastic deformation, such materials no longer behave isotropically.

The question now arises whether in a photoelastic material such as cellulose nitrate, which is suitable for photoplastic models, a breakdown in mechanical coincidence is accompanied by a breakdown in optical coincidence, i.e., whether the optical isoclinic parameters still give the true directions of the principal stresses. To put it another way, the question is whether optical anisotropy is necessarily associated with mechanical anisotropy. This point is of fundamental importance in the application of the shear-difference method to photoplasticity.

The present paper deals with experiments which primarily aim to answer this question. In addition, they throw some light on the general question of mechanical coincidence, in which there is considerable interest today.[2–5]

Mechanical Coincidence

The tests on mechanical coincidence were performed with models loaded in uniaxial tension. The models were made as shown in Fig. 1(a) and the load was applied through friction grips. The model was first subjected to a load of P_1 which was held constant for a sufficiently long time to permit plastic flow to develop. The load was then removed and, after the strain became stabilized, a coupon inclined at angle α to the axis of the model was cut from it, Fig. 1(b). This coupon was then subjected to an axial load P_2 as shown in Fig. 1(c).

In this manner, a state of stress was produced in which the directions of the principal stresses p_1 and p_2 were not constant during the test, the initial direction being determined by P_1 and the final direction by P_2.

The sets of strains produced by P_1 and P_2 will be

M. M. Frocht and R. A. Thomson are associated with Illinois Institute of Technology, Chicago, Ill., and University of Dayton, Dayton, Ohio, respectively.

Paper was presented at 1960 SESA Spring Meeting held in Indianapolis, on May 18–20.

The results presented in this paper were obtained in the course of research sponsored by the Office of Ordnance Research, U. S. Army, under Contract No. DA-11-022-ORD-1609 with Illinois Institute of Technology.

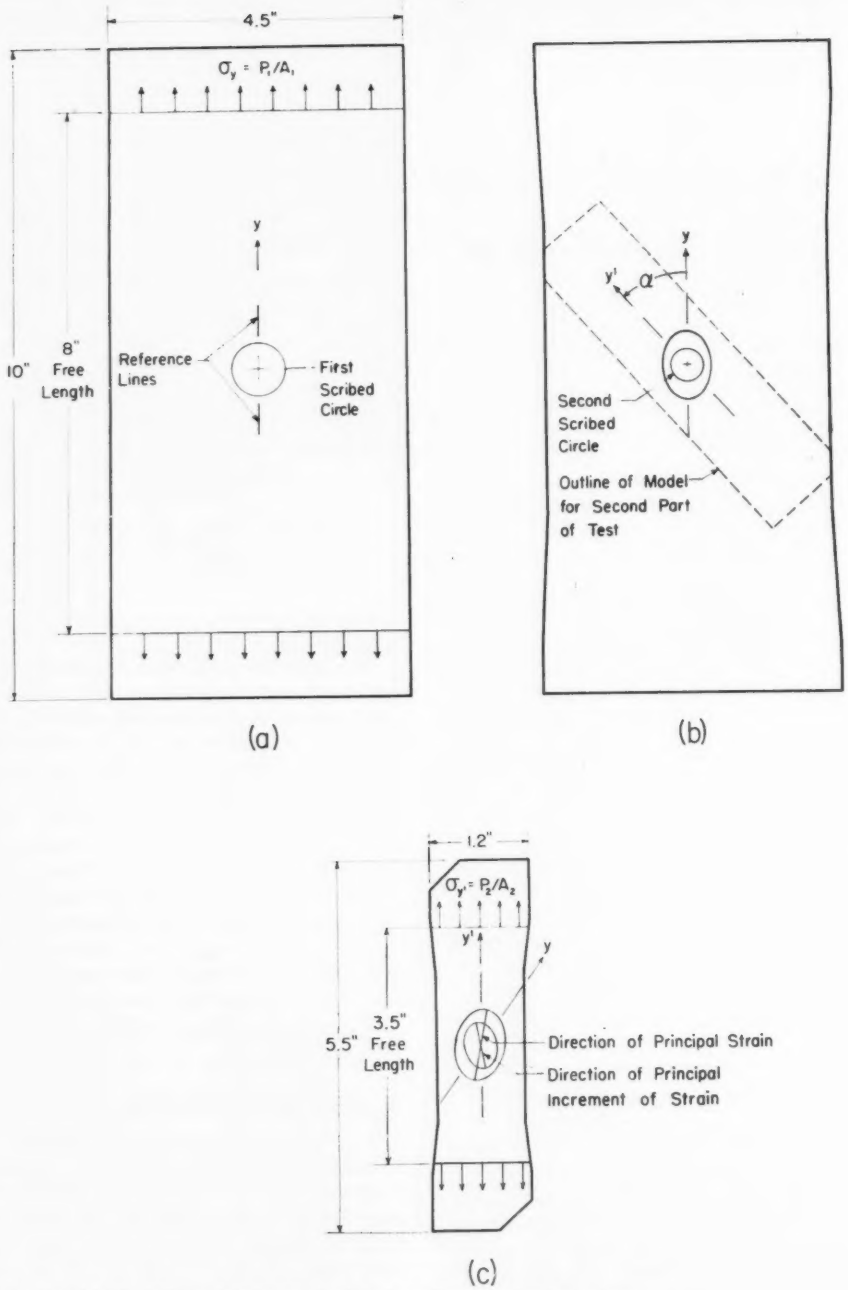


Fig. 1—Models used to determine effects of rotation of principal stresses

referred to as the first and second increments of strain, respectively. The maximum values of these two sets will be referred to as the first and second principal increments of strain, respectively, and will be denoted by $\Delta\epsilon_1$ and $\Delta\epsilon_2$. The resultant maximum strain produced by these increments will be called the total or resultant strain and will be denoted by ϵ_p.

In order to measure the directions of the above strains, a circle approximately 0.6 in. in diameter was scribed on the model before the loads were applied, along with two collinear line-segments to indicate the direction of the initial load P_1. Figure 1(a) shows a sketch of the model immediately after application of load P_1, which was of sufficient magnitude to cause a strain of about 10%. The load P_1 was then removed and sufficient time was allowed for the plastic deformations to recover to a stable state, the amount of plastic recovery being small compared

with the total plastic deformation. When a stable state was attained, the inclined coupon was cut out, and a second circle approximately 0.4 in. in diameter was scribed along with a second pair of collinear line-segments to indicate the direction of the load P_2, Fig. 1(b). This load was large enough to cause an added strain of about the same magnitude as that caused by P_1. Figure 1(c) shows a sketch of the coupon as it appeared after plastic flow had occurred under load P_2. Photographs taken of the model and coupon under these loads were suitably enlarged and the directions of the major axes of the two ellipses were determined. Figure 2 shows photographs of a typical coupon. In Fig. 2(a), load P_1 had been removed but load P_2 had not yet been applied. This photograph shows the relation between p_1 and $\Delta\epsilon_1$. Figure 2(b) is a photograph of the coupon after plastic flow had occurred under P_2, and shows the relation between p_2 and $\Delta\epsilon_2$; as well as between p_2 and ϵ_p. Two angles, ϕ_1 and ϕ_2, were measured. The angle ϕ_1 represents the angle between p_2 and ϵ_p. Similarly, ϕ_2 is the angle between p_2 and $\Delta\epsilon_2$. Table 1 gives the results of the four uniaxial tension tests made.

From Fig. 2 and Table 1, three facts are to be noted. First, Fig. 2(a) shows that $\Delta\epsilon_1$ was in the direction of p_1, i.e., mechanical coincidence exists when the directions of the principal stresses are constant. Second, Fig. 2(b) and Table 1 show that upon changing the directions of the principal stresses following the occurrence of plastic flow, the direction of ϵ_p was not the same as of p_2, i.e., mechanical coincidence does not exist between the resultant strain and the final stress. Third, the directions of $\Delta\epsilon_2$ and p_2 are also different, i.e., mechanical coincidence does not exist between the direction of the second principal increment of strain and the associated increment of stress. This behavior is consistent with that of aluminum.[6, 7]

TABLE 1—RESULTS OF TESTS OF MECHANICAL COINCIDENCE FOR CELLULOID

Model	α, Deg	ϵ_p, %	$\phi_1(p_2,\epsilon_p)$, Deg	$\phi_2(p_2,\Delta\epsilon_2)$, Deg
1	40.7	18.2	19.0	9.8
2	40.7	24.4	7.9	6.4
3	49.9	21.1	16.5	8.6
4	50.7	30.7	15.5	9.3

It should be pointed out that each of the four models used in these tests was cut from the stock sheet with a different orientation. It is assumed that this procedure eliminated major effects due to possible anisotropy in the sheet. If the deviations of ϕ_2 from zero had been due to such anisotropy, the values of ϕ_2 could be expected to be much less consistent than they were found to be.

Optical Coincidence

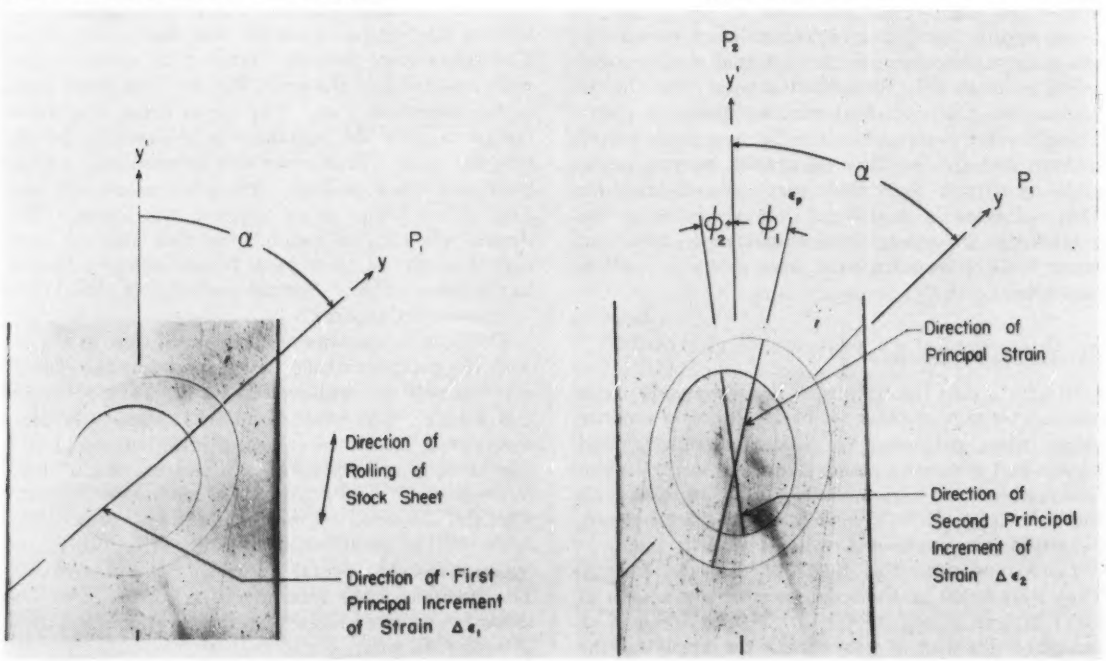
Uniaxial Stresses

During the above uniaxial tests, measurements were also made of the isoclinic parameters at the centers of the ellipses. The measurements showed that

Fig. 2—Photographs of one of the models

(a) Load P_1 removed, and load P_2 not yet applied.

(b) After plastic flow has occurred under load P_2.



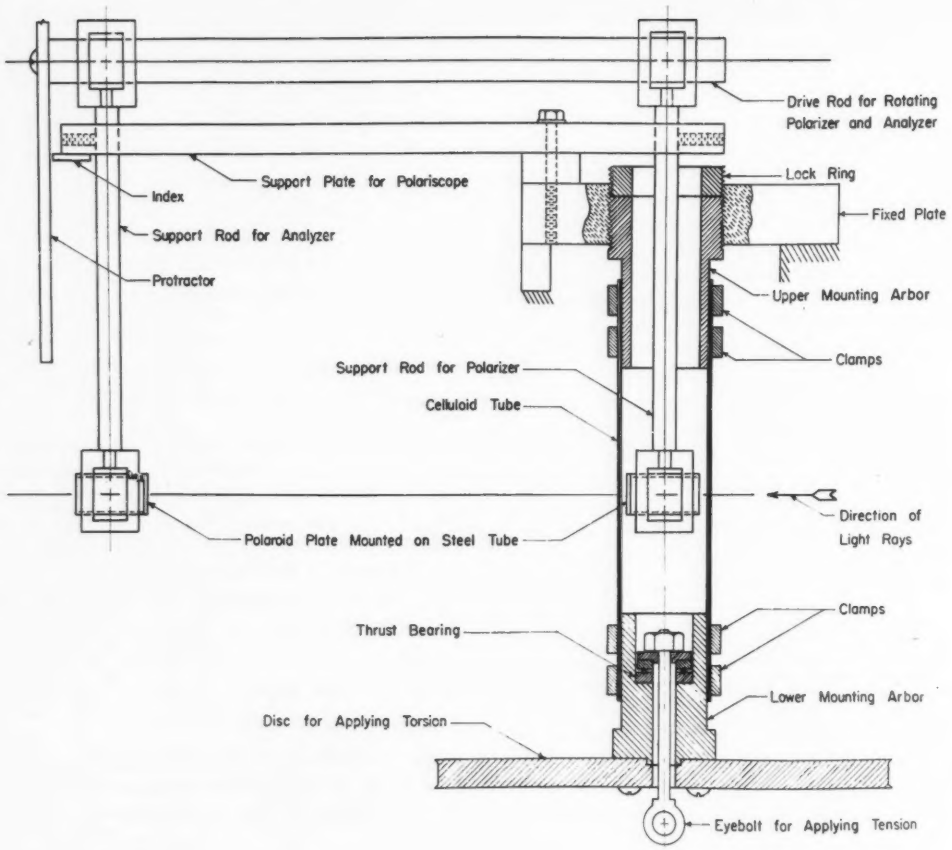


Fig. 3—Loading mechanism and polariscope for tubes in combined tension and torsion

prior to changing the directions of the principal stresses, the isoclinic parameters gave essentially the correct directions of the principal stresses, and failed to do so only for a limited time after the directions of the principal stresses were changed. The difference between the directions of the principal stresses and the isoclinic parameter became negligible as plastic flow took place under load P_2. This behavior is significant. It shows that the breakdown in mechanical coincidence does not carry with it a permanent breakdown in optical coincidence.

Biaxial State of Stresses

In addition to the tests with flat specimens under uniaxial tension, another set of optical tests was run using tubes subjected to combined tension and torsion and to tension alone. In these tests, the isoclinic parameters were measured and compared with the calculated directions of the principal stresses. No strains were measured in these tests.

The tubes were also made of cellulose nitrate. They were 0.040 in. thick, had an inside diameter of 1.457 in. and a length of 7 in. Figure 3 shows an assembly drawing of the fixture for applying the loads in combined tension and torsion and for observing the optical effects in one wall of the tube. The tubes were securely clamped to arbors which were inserted into the ends, Fig. 3. The free length of the tube was 4 in. The upper arbor was made hollow to allow the introduction of a small polarizer into the tube. This arbor was screwed into a fixed plate and was kept from turning by means of a lock ring. The lower arbor carried the loads. The torque was applied through the disk and the axial load through the eyebolt and thrust bearing attached to the lower arbor. Special precautions were taken to assure axial loading.

The polariscope was mounted as shown in Fig. 3, with the polarizer inside the tube so that the optical effect of only one wall was observed. The polarizer and analyzer were made of Polaroid plane-polarizing sheets mounted on the ends of thin-walled steel bushings of the same diameter as the drive rod. These rotate freely in their bearings and are turned in unison with the drive rod by means of endless cords. The drive rod is actuated by turning the protractor which gives the angular position of the polariscope. The isoclinics were determined by means of a photometer and the readings were reproducible within $\pm 1/2$ degree.

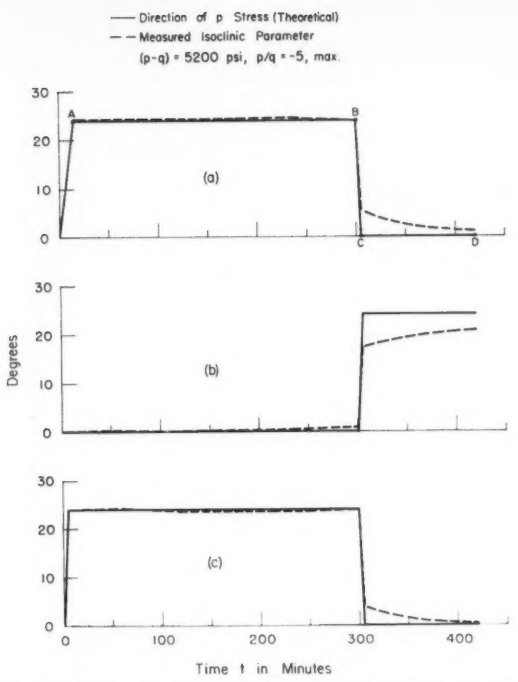


Fig. 4—Effect of rotation of the principal stresses on the isoclinic parameters: (a) and (c) combined tension and torsion followed by pure tension; (b) pure tension followed by combined tension and torsion

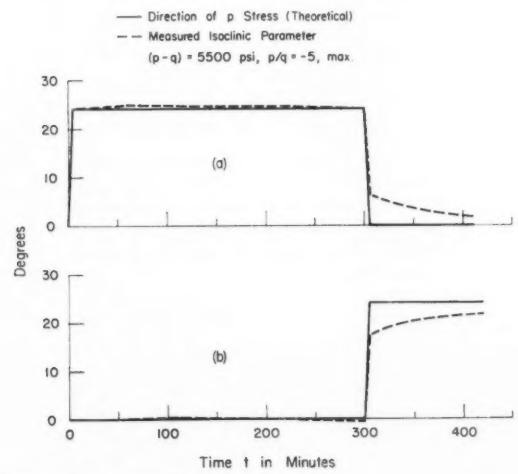


Fig. 5—Same as Fig. 4, except for (p — q)

arbors so as to avoid bending stresses. In order to check the initial alignment of the tube and the load, the isoclinic parameters were measured at the start of the test, and compared with the calculated directions of the principal stresses. No serious deviation was found. It was then assumed that any subsequent development of bending stresses would manifest itself as a disturbance in the observed value of the isoclinic parameter. As long as the parameter indicated the directions of the calculated principal stresses, it was assumed that the alignment was satisfactory. Periodic checks during the tests showed that this condition was fulfilled within ±1 degree.

It was not practicable to make the tubes ideal two-force members by supplying a point suspension at the upper end, due to the necessity of introducing the polarizer through this end. However, any misalignment in loading could be readily detected by inspection of the stress pattern and corrected. Precautions were taken in aligning the tubes on the

Five tests in combined tension and torsion were conducted. In three of these tests, Fig. 4, the tubes were loaded to a value of $(p - q)$ equal to 5200 psi, with the ratio p/q equal to -5 in combined tension and torsion. In the other two tests, Fig. 5 $(p - q)$ was made equal to 5500 psi, with the same value of p/q. The ratio of p to q used was the largest which the tubes could withstand without buckling. The first stress system was held constant for 5 hr during which time a plastic deformation of 2 to 3% was developed. During this time, periodic measurements of the isoclinic parameters were made. The stress system was then changed to one having different directions of the principal stresses, but the same value of $(p - q)$. This was done in order to compare the birefringence (not discussed in this paper) and also to insure that the tubes remained in the plastic state. Also, in order to prevent buckling, the second stress system was allowed to act during an interval of time in which the strain did not exceed 5%. The isoclinic parameter was again measured periodically. Figures 4 and 5 show the order in which the loads were applied, the theoretical directions of the p stress and the variations in the isoclinic parameters which were measured from the axis of the tube.

The results thus show that, under a biaxial state of stress with constant directions of the principal stresses, the isoclinic parameter gives the directions of the principal stresses. Moreover, the results show that it is possible to have optical isotropy in the presence of mechanical anisotropy, i.e., even when the directions of the principal stresses are not constant, the isoclinic parameters give the directions of the principal stresses, except for a period of about 2 hr immediately after changing the direction of the stresses.

Further work on this problem is being planned.

Bibliography

1. *Frocht, M. M., and Thomson, R. A., "Studies in Photoplasticity," Proceedings Third United States National Congress of Applied Mechanics, p. 533, June 1958.*
2. *Drucker, D. C., "Relation of Experiments to Mathematical Theories of Plasticity," Journal of Applied Mechanics 16 (4), 349 (1949).*
3. *Davis, E. A., "Combined Tension-Torsion Tests With Fixed Principal Directions," Journal of Applied Mechanics 22 (3), 411 (1955).*
4. *Phillips, Aris, and Kaechele, Lloyd, "Combined Stress Tests in Plasticity," Journal of Applied Mechanics 23 (1), 43 (1956).*
5. *Marin, Joseph, Ulrich, B. H., and Hughes, W. P., "Plastic Stress-Strain Relations for 75S-T6 Aluminum Alloy Subjected to Biaxial Tensile Stresses," NACA Technical Note 2425, August 1951.*
6. *Naghdi, P. M., Essenberg, and Kopf, "Initial and Subsequent Yield Surfaces in Plasticity," Journal of Applied Mechanics 25, 201 (June 1958).*
7. *Hu and Bratt, "Effect of Tensile Plastic Deformation on Yield Condition," Journal of Applied Mechanics 25, 411 (September 1958).*

Experiments on Damping at Contacts of a Sphere with Flat Plates

Test results for the constant normal force/varying tangential force problem are compared with theoretical predictions

by L. E. Goodman and G. E. Bowie

ABSTRACT—An apparatus for studying slip damping at rounded contacts for three different systems of forces is described. Experimental results for the constant normal force/varying tangential force problem when a Type 316 stainless-steel sphere is driven between two flat Type 316 stainless-steel plates are presented. They are in fair agreement with theoretical predictions by R. D. Mindlin, *et al.*

L. E. Goodman and G. E. Bowie are Professor of Mechanics and Research Fellow, respectively, Department of Aeronautical Engineering, University of Minnesota, Minneapolis.

Paper was presented at 1960 SESA Annual Meeting held in Berkeley, Calif., on October 19–21.

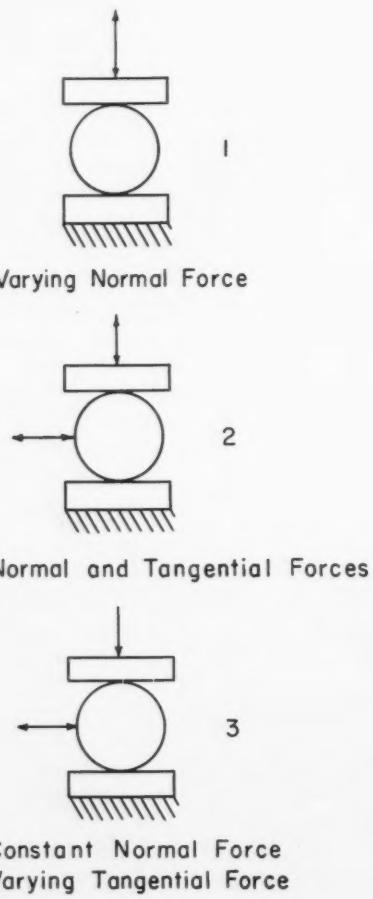
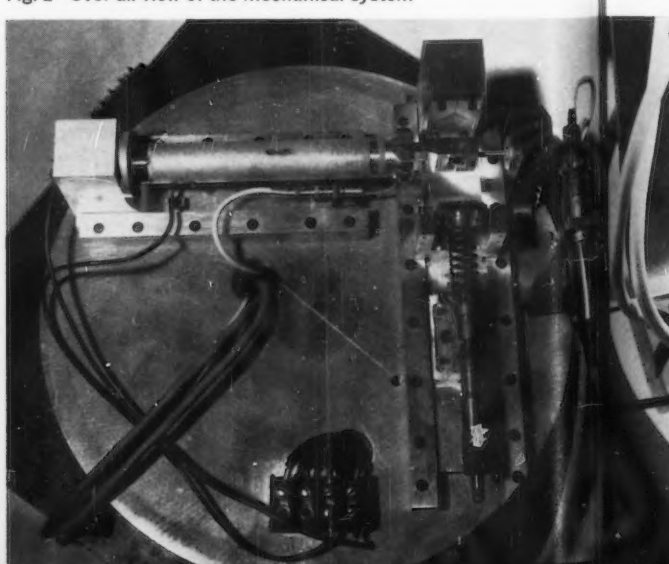


Fig. 1—Loading conditions for three different contact problems

List of Symbols

- a = outer contact radius, in.
- c = inner contact radius, in.
- E = modulus of elasticity, psi.
- f = static coefficient of friction.
- G = shearing modulus, psi.
- N = normal force, lb.
- T = tangential force, lb.
- T^* = maximum tangential force during a cycle, lb.
- T_{max} = tangential force at the onset of gross slip, lb.
- X_p = component of traction in the direction of applied T, psi.
- Z_p = component of traction normal to a contact surface, psi.
- δ = tangential displacement, in.
- δ^* = maximum tangential displacement during a cycle, in.
- δ_{max} = tangential displacement at the onset of gross slip, in.
- δ_i = tangential displacement for T increasing, in.
- δ_d = tangential displacement for T decreasing, in.
- ΔE = energy dissipation per cycle, in-lb/cycle.
- μ = Poisson's ratio.

Fig. 2—Over-all view of the mechanical system



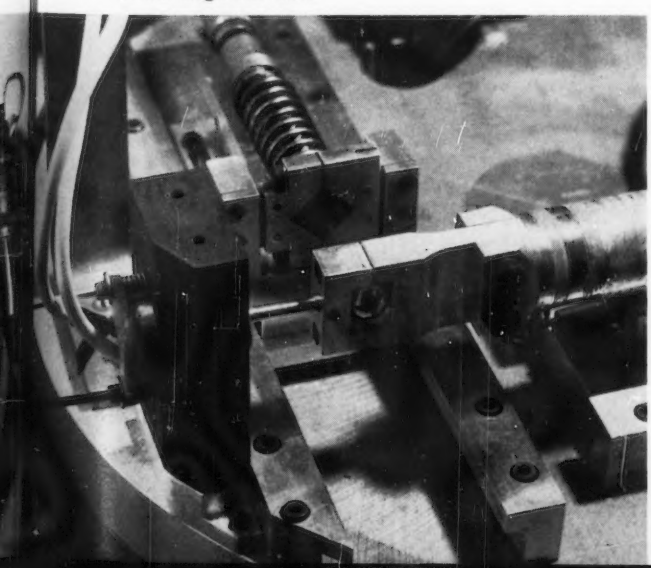
Introduction

An apparatus was constructed for studying contact problems under the conditions shown in Fig. 1. The energy dissipation per cycle during oscillatory loading is of principal interest here. Results for the constant normal force/varying tangential force problem are compared herein with theoretical predictions. When relative movement occurs between some, but not all, of the particles on either side of a contact surface, it is said that "no-gross-slip" conditions prevail but that "microscopic sliding" takes place. The terms "macroscopic sliding," "gross slip" and "slide" apply when relative movement takes place between all particles on either side of the contact. It is the energy dissipated at tangential loads less than those required to produce gross slip which has been of primary interest to us. We have found that, as gross slip is approached in experiments at a frequency of 60 cps, the tangential force does not behave in the manner anticipated by the theory for a quasi-static model. Nevertheless, a comparison of theoretical and experimental energy dissipation per cycle can be made when the experimental hysteresis loop corresponding to the theoretical loop for the onset of gross slip is identified.

Description of the Apparatus

The constant normal force/varying tangential force problem has been studied with two other experimental arrangements. The design of our apparatus is similar to that of T. F. Osmer and W. P. Mason.[1] We copied their use of a hollow cylinder of barium titanate as an electrostrictive driving transducer, an illuminating-tube transformer as a driving voltage source at line frequency, and a disk of barium titanate cemented between the driving transducer and the sphere to measure force. A quite different design used by K. L. Johnson[2] is well suited for static experiments. He imbedded three spheres near the periphery of a disk. The spheres protruded so that they could be placed in contact with three flat surfaces. A torsional pendulum was attached to the disk containing the spheres. Moments of the forces at the three contacts resisted torque applied by the pendulum. Johnson performed dynamic experiments by exciting the torsional pendulum at its resonant frequency. As gross slip was approached, he found that the amplitude of the vibrating system became violently unsteady. In our experiments, the driving transducer is essentially a displacement generator which produces sinusoidal tangential displacements of a sphere pressed between two parallel flat plates for conditions both above and below gross slip.

Fig. 3—Arrangement for attaching a $^1/_2$-in. diam sphere to the tangential driver



Photographs of the mechanical system are shown in Figs. 2 and 3. In Fig. 3 a $^1/_2$-in. diam sphere is rigidly coupled to an electrostrictive tangential driver. Two flat specimens of the same material as the sphere are brought into contact with the sphere during experiments. One of the flat specimens is rigidly attached to a block while the other is held in a flex-plate assembly. Normal force is applied by the flats to the sphere when a spring is advanced against the flat specimen held in the holder mounted on flex-plates.

A second electrostrictive transducer, cemented between the main driving element and the sphere holder, senses force exerted at the contacts. The core of a linear-variable-differential transformer is coupled to the end of the ball holder for displacement measurements. An over-all schematic arrangement of the force- and displacement-measuring systems is shown in Fig. 4. Electrical measures of force and displacement are placed on the horizontal and vertical inputs of an oscilloscope to obtain a hysteresis loop whose area is a measure of energy dissipation per cycle at the two contacts of the sphere with the flat specimens.

The force- and displacement-measuring systems were calibrated at a frequency of 60 cps with the aid of a Tuckerman optical extensometer and autocollimator. By replacing the conventional tungsten lamp source in the Tuckerman autocollimator with a stroboscopic source synchronized with the motion of the driving transducer, it was possible to obtain a single reading of displacement within 2 microinches. The stroboscopic source is described in the Appendix.

When force and displacement signals are placed on horizontal and vertical plates of an oscilloscope, the resulting loop area on the oscilloscope screen is, of course, only an accurate measure of energy dissipation at the contacts of the sphere if phase shifts introduced electrically are properly compensated. To accomplish this, we place an axial spring-type load on the end of the tangential driver and thus obtain a set of conditions in which the mechanical energy dissipated is negligible in comparison with the expected energy loss in the contact problem under consideration. We then adjust a phase shifter in the displacement-measuring system so that the force-displacement relation on the oscilloscope is one of

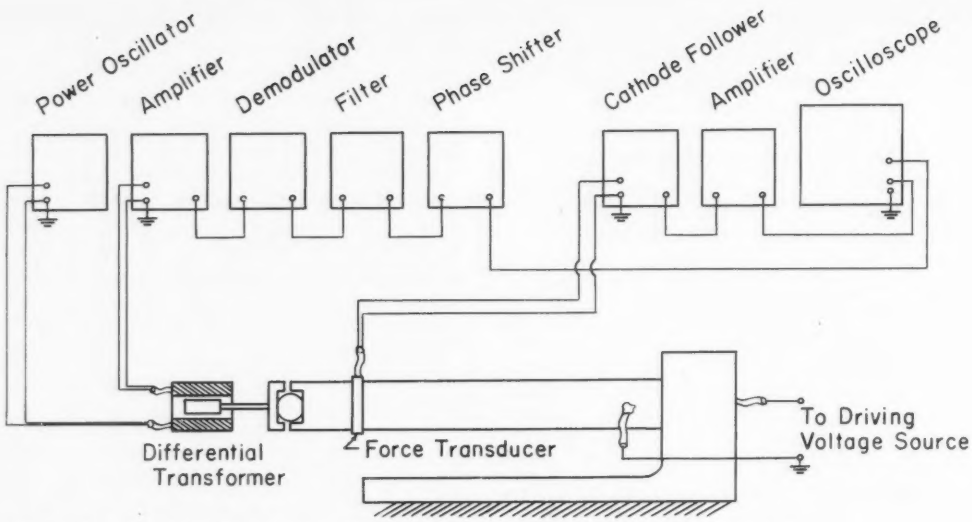


Fig. 4—Over-all schematic arrangement of the force and displacement-measuring systems

zero area. Detailed descriptions of the apparatus and experimental procedures are given in Reference 3.

Theoretical Predictions

An extensive theoretical treatment of the constant normal force/varying tangential force problem has been published.[1] The principal investigator is R. D. Mindlin, who considers the problem within the context of the classical theory of elasticity. Mindlin determined the distribution of the component of traction X_p, in the direction of an applied tangential force T, on the circular contact area between two elastic spheres. He found that the condition of no relative slip between points on either side of the contact requires that X_p be singular at the boundary of the contact region. He therefore concluded that slip must occur in practice when a tangential force is applied to two bodies in contact, and that it begins at the contact boundary.[4]

The classical Hertz theory of contact stresses[5] predicts the normal component of traction Z_p which occurs when two lubricated spheres are pressed together with a normal force N. Mindlin assumed that, when the spheres are not smooth and a tangential force is applied, the tangential component of traction X_p at any point on the contact surface cannot exceed the product of the static coefficient of friction f and Z_p. Thus, when a tangential force T (less than fN) is applied, slip penetrates from the outer contact radius a to an inner radius c. It is assumed that X_p remains at its greatest possible value fZ_p within the included annulus. In his solution, Mindlin found a distribution of X_p which produces displacements satisfying a no-slip condition within the circle of radius c. He found that this radius is given by the expression

$$\frac{c}{a} = \left(1 - \frac{T}{fN}\right)^{1/3} \tag{1}$$

and that the tangential displacement relative to one another of distant points on opposite sides of the contact is

$$\delta = \frac{3(2-\mu)}{8Ga} fN \left[1 - \left(1 - \frac{T}{fN}\right)^{2/3}\right] \tag{2}$$

where μ is Poisson's ratio and G the shear modulus.

The theory was extended (excluding inertia effects) to the case in which T is increased from zero to $T^* \leq fN$; reduced to $T = -T^*$; and again increased to $T = T^*$.[4] In this way, a closed force-displacement hysteresis loop is obtained. Its area represents the energy loss per cycle, ΔE, due to friction on the annulus over which slip occurs. If δ_d represents the displacement of distant points on opposite sides of the contact for T decreasing from T^* to $-T^*$, and δ_i for T increasing from $-T^*$ to T^*, then

$$\delta_d(T) = \frac{3(2-\mu)}{8Ga} fN \left[2\left(1 - \frac{T^* - T}{2fN}\right)^{2/3} - \left(1 - \frac{T^*}{fN}\right)^{2/3} - 1\right] \tag{3}$$

and

$$\delta_i(T) = -\delta_d(-T) \tag{4}$$

The energy loss per cycle due to friction on the annulus is given by the expression:

$$\Delta E = \frac{9(2-\mu)}{5Ga} (fN)^2 \left\{1 - \left(1 - \frac{T^*}{fN}\right)^{5/3} - \frac{5T^*}{6fN}\left[1 + \left(1 - \frac{T^*}{fN}\right)^{2/3}\right]\right\} \tag{5}$$

Comparison of Experimental and Theoretical Hysteresis Loops

The results presented here are for a $^1/_2$-in. diam sphere pressed between two $^1/_2$-in.-square by $^1/_4$-in. thick plates. All three specimens were made of Type 316 stainless steel. They were handled with rubber gloves and cleaned with optical tissue moistened in trichlorethane.

We learned, by examining tangential-force wave

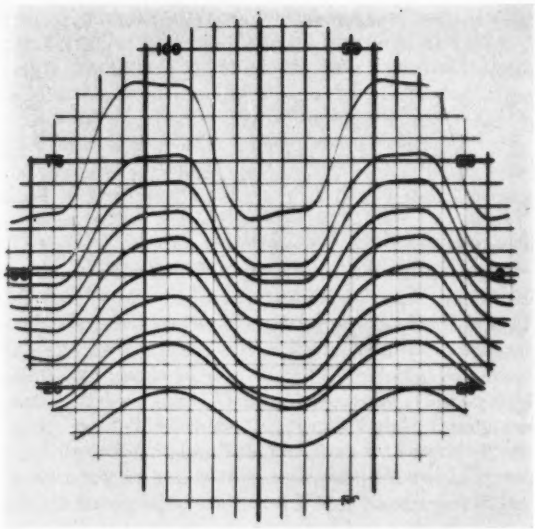


Fig. 5—Tangential-force wave forms for a normal force N = 0.24 lb

forms for a given normal force N on an oscilloscope screen, that changes in driving voltage from low values which do not cause slip over the entire contact region to higher values which do cause gross slip are accompanied by definite changes in the shape of the *force* wave forms. A variable band-pass filter was used to analyze the wave forms. It was found that the *displacement* wave form consisted of 60 and 180 cps components with the ratio of the amplitude of the higher frequency component to that of the lower, being no larger than 0.04, even for gross-slip conditions. This ratio is so small that the displacement wave forms may be regarded as sinusoidal. They so appear when viewed on an oscilloscope screen. We therefore refer to the barium titanate driving transducer as a displacement generator and say that it produces sinusoidal displacements for either gross-slip or no-gross-slip conditions.

Now consider a typical set of tangential-*force* wave forms in the neighborhood of gross slip. In Fig. 5, ten such force wave forms for a normal force $N = 0.24$ lb are shown. In what follows, they are referred to as waves 1 to 10, numbering from the bottom of the illustration. They correspond to ten peak displacements which increase by increments, the smallest displacement corresponding to wave 1 and the largest to wave 10. Displacements smaller than that corresponding to wave 1, although not shown here, are associated with sinusoidal, force wave forms. Wave 1 is still sinusoidal, but, for higher displacements, the force wave forms (waves 2–10) are no longer sinusoidal. It is instructive to try to select from among these wave forms the one which most nearly seems to separate waves for gross-slip conditions from waves for no-gross-slip conditions. One may attempt the selection on the basis of a preconceived idea of what the shape of the wave should be. Intuitively, one might idealize the conditions at gross slip by thinking in terms of a displacement generator driving a spring which is in series with a Coulomb damping device. When displacements are large enough to cause slip in the Coulomb device, the wave form of the force exerted on the system by the generator will look like a peak-clipped sine wave. With a criterion based on this model, wave 8 in Fig. 5 would be selected as corresponding to the onset of gross slip. This choice then yields a value for T_{max} and, therefore, a value for f. This value of f substituted in eq 5 together with ΔE taken from the area of the corresponding hysteresis loop permits computation of the factor $9(2 - \mu)/5Ga$ appearing in eq 5. All hysteresis loops for lesser values of T^* can then be compared with the predictions of the theory. There is, of course, a perfect matching at $T^* = T_{max}$ when the comparison is made in this way. As an alternative, one might compute the factor $9(2 - \mu)/5Ga$ from a knowledge of the properties of the material, and thus avoid an enforced match of theory and experiment at $T^* = T_{max}$. These procedures lead, however, to an unsatisfactory comparison between theory and experiment. As a matter of fact, the T_{max} corresponding to the onset of gross slip is very nearly that associated with wave 4 and not that associated with wave 8.

Instead of looking for the first flat-topped force wave, one might expect, with the theory of Mindlin, *et al.*, in mind, that below gross slip a single value of peak force would correspond to a given peak displacement. As the peak displacement is increased in a series of experiments, the peak force would be expected to increase to a limiting value determined by the product fN of a coefficient of friction f and the normal force N. This approach to the determination of T_{max} and the corresponding ΔE encounters grave difficulty, however, because the peak magnitude of the force wave form changes very little in the neighborhood of gross slip, while the energy dissipation changes considerably. Therefore, small differences in estimations of f cause large differences in estimates of the energy dissipation at the onset of gross slip.

The foregoing paragraphs are intended to describe the main difficulties which arise when an attempt is made to compare theory with experiment in this field. These difficulties are revealed only when the force-time variation is actually measured. In order to arrive at a satisfactory basis for comparison of experimental results with theoretical predictions we select, in the way described below, from among the experimental hysteresis loops near the onset of gross slip, the one which most nearly matches the theoretical loop for onset of gross slip. We find a close agreement in size and the shape between one of our experimental loops and the theoretical loop for the onset of gross slip. We can then test the theory by determining whether or not the agreement in size and shape holds for smaller loops representing conditions below gross slip.

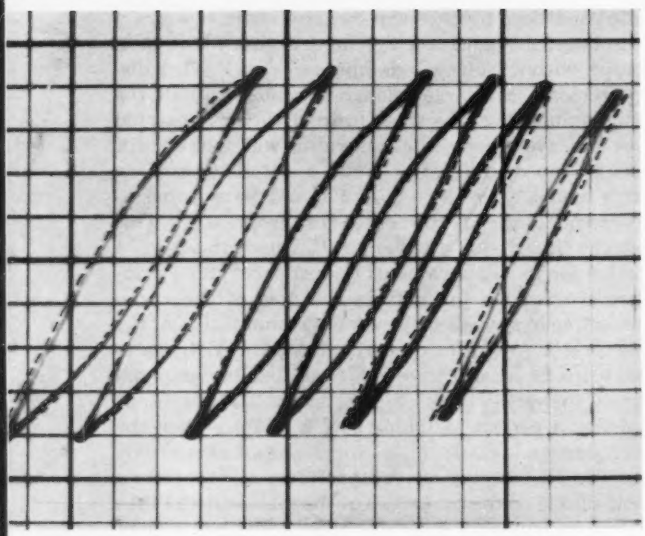


Fig. 6—Dotted theoretical loops superposed on solid experimental loops. Normal force N = 0.24 lb. Ordinate: tangential force, 1 scale unit ≡ 0.115 lb. Abscissa: tangential displacement, 1 scale unit ≡ 4.48 microin.

Peak-to-peak magnitudes of wave forms are always measured in the experiments. To simplify the notation we may say that we measure δ^*, T^* and ΔE for a given loop when a sphere is in single contact even though we actually measure $2\delta^*$, $4T^*$ and $2\Delta E$ for a sphere in double contact with flat plates. Theoretical hysteresis loops may be plotted using eqs 3 and 4 in the dimensionless forms:

$$\frac{\delta_d}{\delta_{max}} = 2\left(1 - \frac{T^* - T}{2T_{max}}\right)^{2/3} - \left(1 - \frac{T^*}{T_{max}}\right)^{2/3} - 1 \quad (6)$$

$$\frac{\delta_i}{\delta_{max}}(T) = -\frac{\delta_d}{\delta_{max}}(-T) \quad (7)$$

When $T = 0$ and $T^* = T_{max}$,

$$\delta_d/\delta_{max} = 2(1 - {}^1/_2)^{2/3} - 1 = 0.25992$$

From eqs 6 and 7, it follows that when $T = 0$ and $T^* = T_{max}$, $(\delta_d - \delta_i)/2\delta_{max} = 0.25992$

In an experiment at a given normal load, we seek the hysteresis loop having the property that when $T = 0$,

$$(\delta_d - \delta_i)/2\delta^* = 0.260 \quad (8)$$

For this loop we say that δ^* and T^* are equivalent to δ_{max} and T_{max}, respectively. We subsequently lower the driving voltage to obtain loops of smaller area. (A practical detail of technique arises when oscilloscope jitter makes it difficult to find loops satisfying eq 8 exactly. We then interpolate by assuming that the theoretical relations

$$\frac{\delta^*}{\delta_{max}} = 1 - \left(1 - \frac{T^*}{T_{max}}\right)^{2/3} \quad (9)$$

and

$$\left[\frac{\delta_d - \delta_i}{2\delta_{max}}\right]_{T=0} = 2\left(1 - \frac{T^*}{2T_{max}}\right)^{2/3} - \left(1 - \frac{T^*}{T_{max}}\right)^{2/3} - 1 \quad (10)$$

hold in the experiments when δ^* is near δ_{max}. On dividing eq 10 by eq 9 the following equation results:

$$\left[\frac{\delta_d - \delta_i}{2\delta^*}\right]_{T=0} = \frac{2\left(1 - \frac{T^*}{2T_{max}}\right)^{2/3} - \left(1 - \frac{T^*}{T_{max}}\right)^{2/3} - 1}{1 - \left(1 - \frac{T^*}{T_{max}}\right)^{2/3}} \quad (11)$$

The effect of the interpolation is minor.)

For example, the solid loops in Fig. 6 were obtained in an experiment with a normal force $N = 0.24$ lb. The ordinate is tangential force and the abscissa tangential displacement. For the loop on the extreme left, $[(\delta_d - \delta_i)/2\delta^*]_{T=0}$ is equal to 0.25. Using eq 11, we found that $T^*/T_{max} = 0.998$ and, using eq 9, that $\delta^*/\delta_{max} = 0.984$ for this loop. Figure 6 shows how experimental and theoretical loops compare in shape. The dotted loops were computed from eqs 6 and 7 and were superposed on the solid experimental loops to the scale of the experimental loops.

Equation 5, for the energy loss per cycle, may be written in the dimensionless form:

$$\frac{\Delta E}{\delta_{max} T_{max}} = \frac{24}{5}\left\{1 - \left(1 - \frac{T^*}{T_{max}}\right)^{5/3} - \frac{5T^*}{6T_{max}}\left[1 + \left(1 - \frac{T^*}{T_{max}}\right)^{2/3}\right]\right\} \quad (12)$$

K. L. Johnson used instead the dimensionless form:

$$\frac{Ga\Delta E}{T^{*2}} = \frac{9(2 - \mu)}{5}\left(\frac{T_{max}}{T^*}\right)^2\left\{1 - \left(1 - \frac{T^*}{T_{max}}\right)^{5/3} - \frac{5T^*}{6T_{max}}\left[1 + \left(1 - \frac{T^*}{T_{max}}\right)^{2/3}\right]\right\} \quad (13)$$

He experimentally determined a value of the coefficient of friction f and obtained a theoretical curve of $Ga\Delta E/T^{*2}$ vs. T^*/N using this value of f. In reducing his experimental measurements of energy loss per cycle to the same form, he assumed values of the elastic constants, G and E, and computed radii of contact regions using the following relation predicted by the Hertz theory[5]:

$$a = 1.11\left(\frac{NR}{E}\right)^{1/3}$$

where R is the radius of a sphere in contact with a flat surface.

As indicated above, experimental energy dissipation near the onset of gross slip is a multiple-valued function of T^*/T_{max} while it is a single-valued function of δ^*/δ_{max}. It follows that δ^*/δ_{max} is a better choice of parameter than T^*/T_{max} for comparing theory and experiments. The solid theoretical curve of $\Delta E/\delta_{max}T_{max}$ vs. δ^*/δ_{max} in Fig. 7 was obtained from eqs 9 and 12. The experimental points of Fig. 7 were obtained in three different tests corresponding to three different values of normal force N.

Discussion

The theoretical curve of $\Delta E/\delta_{max}T_{max}$ vs. δ^*/δ_{max} in Fig. 7 applies to any two elastic, homogeneous, iso-

tropic, smooth spheres with identical elastic constants for any values of normal force N and microscopic coefficient of static friction f as long as δ^* is less than δ_{max}, and as long as hysteresis loops are traversed quasi-statically. We have compared this theoretical curve with experimental results obtained at a frequency of 60 cps using Type 316 stainless-steel specimens. As shown in Fig. 7, there is fair agreement between the theoretical curve and experimental values of $\Delta E/\delta_{max}T_{max}$ for three different values of N when δ^*/δ_{max} is in the range 0.5 to 1.0. When δ^*/δ_{max} is less than 0.5, it is difficult to compare experimental and theoretical values of energy dissipation because of limitations in the present method of measuring energy dissipation.

The experimental loop on the extreme left in Fig. 6 was matched with a theoretical loop for conditions near the onset of gross slip. One can see that the agreement in shape of these two loops is maintained for the remaining pairs of loops obtained at successively lower values of peak displacement. This is a more sensitive test of the theory than is provided by the graphs of Fig. 7. It can be concluded from examination of Fig. 6 that the theoretical analysis of the mathematical model for the problem considered accurately predicts the shape of experimental hysteresis loops obtained when using Type 316 stainless-steel specimens. To the best of our knowledge, this is the first time experimentally determined dynamic hysteresis loops have been shown to conform to the shape predicted by theory.

We have described how the tangential-force wave form changes shape while the displacement wave form remains sinusoidal as conditions change from no-gross slip to gross slip. It is possible that similar behavior takes place for constant normal force/varying tangential force problems having geometries other than provided by the contact of spheres. For example, one may compare our force wave forms of Fig. 5 with those obtained by Mason and White[6] in a study of wear in which a displacement generator drove a piece of wire in contact with a flat specimen at a frequency of 18,000 cps. Here, too, force wave forms appear sinusoidal well below gross slip and then progressively change toward waves with flat tops as the peak displacement is increased.

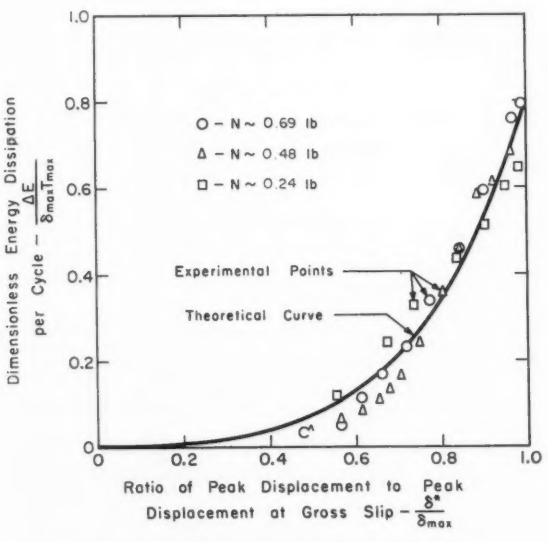


Fig. 7—Dimensionless energy dissipation $\Delta E/\delta_{max}T_{max}$ versus ratio of peak displacement to peak displacement at gross slip δ^*/δ_{max}

In Reference 3, we compare experimental results for Type 316 stainless-steel specimens with those for pyrex glass. With stainless-steel specimens under a constant normal load, one may cycle the peak displacement and reproduce loops quite well. A loop for a given peak displacement will maintain its size and shape on the oscilloscope screen for tens of minutes. However, with pyrex-glass specimens, loops may change almost instantaneously or change over periods of minutes even though the operator does not disturb the system. Differences such as those that we have observed between the behavior of Type 316 stainless steel and pyrex-glass specimens at a frequency of 60 cps must be taken into account in any attempt to apply the theory of Mindlin, *et al.*, to a wide class of materials.

APPENDIX

A Stroboscopic Light Source for a Tuckerman Autocollimator

The force-measuring transducer used in these experiments is an electrostrictive element whose impedance is largely capacitive. Consequently, it is desirable to calibrate it dynamically at the working frequency of 60 cps. The circuitry for the displacement-measuring transducer and the mechanical arrangement of the transducer also make dynamic displacement calibration desirable.

The peak-to-peak displacements involved in our experiments are less than 100 microinches. A Tuckerman optical extensometer was selected as a displacement standard.[7] A Tuckerman unit consists of an autocollimator and an optical extensometer. Conventionally, a tungsten lamp is used as a light source for the autocollimator. It may be used to illuminate either of two types of fiduciary marks. The mark designed for static calibration has a reading hairline and a vernier while the mark for dynamic work is dumbbell-shaped.[8] Inherently, greater precision may be obtained with the former type. The sensitivity of the autocollimator with the vernier-type mark is 2 microinches. We obtain this sensitivity in dynamic measurements by retaining the vernier-type mark and by using a stroboscopic light source.

The physical arrangement of the new source and the associated electrical circuitry are shown in Fig. 8. A $^1/_4$-in. diam hole was drilled in the side of the autocollimator near the prism which directs light

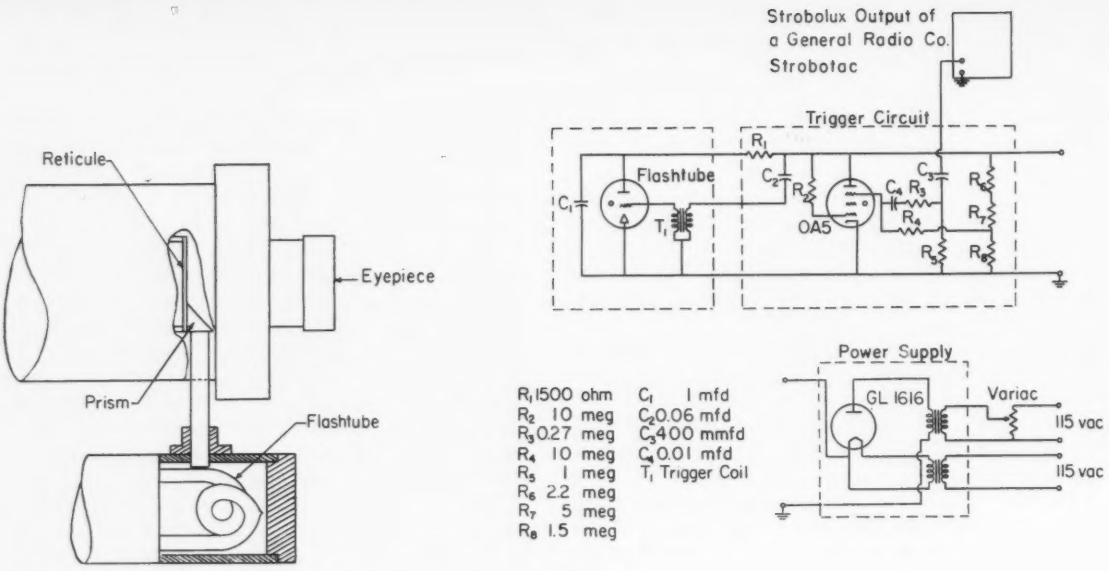


Fig. 8—Stroboscopic light source for a Tuckerman autocollimator

onto the fiduciary mark. A rod of plexiglass with optically polished ends couples the light from a flashtube[9] to the prism. In the electrical circuit, a OA5 trigger tube is triggered by the Strobolux output pulses of a General Radio Co. Strobotac. The trigger tube output is placed on the grid of the flashtube. When the strobotac is operated in its line-frequency position, the trigger pulses are synchronized with the driving voltage of the tangential driver whose displacement is to be measured. By adjusting the variac in the circuit shown, the power-supply voltage may be increased until the main discharge capacitor C_1 is charged to a voltage which is high enough to cause the flashtube to fire. A repetition rate of 15, 30 or 60 flashes per second, depending on the power-supply voltage, takes place in synchronism with the mechanical motion.

A driving-voltage source of continuously variable phase is used when calibrating the apparatus. In the source, three wye-delta connected transformers supply the three stator coils of a selsyn motor from three-phase line terminals. A clamping arrangement allows one to position the rotor at any desired angular position by observing the reading of a pointer attached to the shaft of the rotor against a fixed, graduated circle. A single-phase output voltage taken from the rotor of the selsyn varies in phase when the rotor shaft is rotated. A voltage divider feeds a fraction of the selsyn output into a power amplifier which is connected to the low-voltage winding of an illuminating-tube transformer. The high-voltage winding provides the excitation voltage for the barium titanate driving transducer. A driving voltage of desired magnitude is obtained by adjusting the voltage divider.

Peak-to-peak displacements of the sphere holder shown in Fig. 3 are measured in both displacement and force calibrations by a Tuckerman optical extensometer. When the continuously variable phase shifter in the power supply for the barium titanate driver used in the experiments is slowly adjusted, one can see the image of the fiduciary mark move on the reticule of the autocollimator. In this way, the position of the phase shifter for a maximum reading of the Tuckerman unit is obtained. A minimum reading is then found by making a phase shift of 180 deg in the power supply. With this technique, readings are taken when the velocity of the motion produced by the driver is practically zero. Thus, errors that might arise because of finite flash duration are minimized.

For force calibrations, two dynamometers of different stiffnesses are used. Each is shaped like a fixed-ended beam. The free end of the sphere holder is brought into contact with a beam at mid-span and the displacements of the beam corresponding to force measuring system outputs for different driving voltages are noted. Beam stiffnesses are obtained in static calibrations using dead weights.

Acknowledgment

The work described in this report was part of a continuing program supported by the Materials Laboratory, Wright Air Development Division.

Bibliography

1. *Mindlin, R. D., Mason, W. D., Osmer, T. F., and Deresiewiez, K., "Effects of an Oscillating Tangential Force on the Contact Surfaces of Elastic Spheres,"* Proc., First U. S. Natl. Congr. Appl. Mech., *203–208 (1951).*

2. *Johnson, K. L., "Surface Interaction Between Two Elastically Loaded Bodies Under Tangential Forces,"* Proc., Roy. Soc. London, *Ser. A, 230, 531–548 (1955).*

3. *Goodman, L. E., and Bowie, G. E., "Experiments on Slip Damping at Rounded Contacts," Wright Air Development Division Technical Rept. 60–161.*

4. *Mindlin, R. D., "Compliance of Elastic Bodies in Contact,"* Jnl. Appl. Mech., *16, 259–268 (1949).*

5. *Timoshenko, S., and Goodier, J. N.,* Theory of Elasticity, *McGraw-Hill Book Co., Inc., New York, 2nd ed., 372–377 (1951).*

6. *Mason, W. P., and White, S. D., "New Techniques for Measuring Forces and Wear in Telephone Switching Apparatus,"* Bell System Tech. Jnl., *31, pp. 469–503 (1952).*

7. *Tuckerman, L. C., "Optical Strain Gages and Extensometers,"* Proc. Am. Soc. Testing Matls., *Part 2, 23, 602–610 (1923).*

8. Tuckerman Optical Strain Gages, *American Instrument Co., Inc., Silver Spring, Md., Bull. 2084, 1–8, 1948.*

9. Flashtubes, *General Electric Lamp Division, Cleveland, Application Engineering Bull. LS-110, 1–8 (1954).*

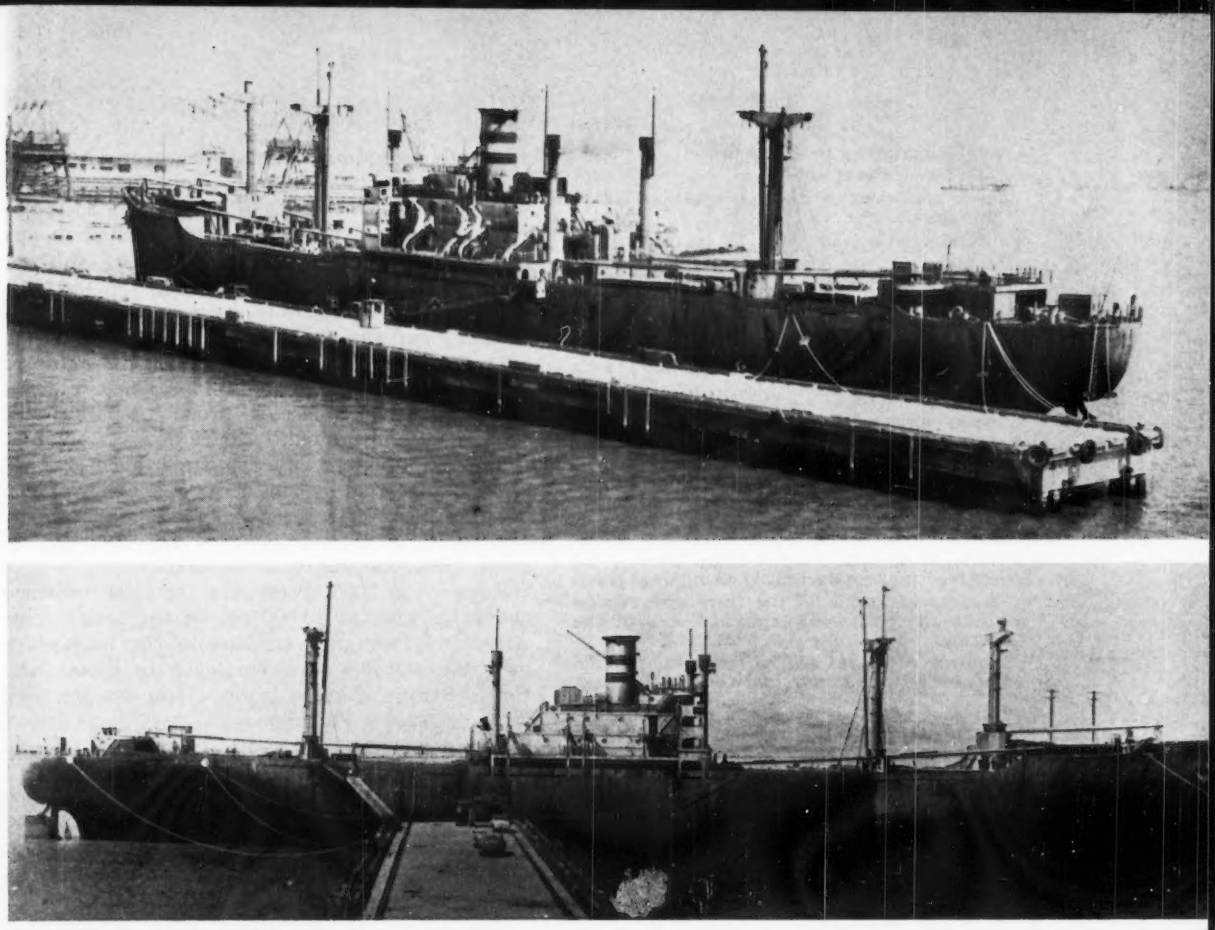


Fig. 1—Ship position for Test II (top) and Test III (bottom)

Measurement of Thermal Stresses in the S.S. "Boulder Victory"

Experimental study provides reliable prototype measurements of thermal-stress patterns around a complete transverse section of a ship

by J. L. Meriam, R. F. Steidel, G. W. Brown and P. T. Lyman

ABSTRACT—A lack of comprehensive experimental measurements of thermal stresses induced in a ship's hull structure by diurnal temperature changes prompted this study. Its essential purpose was to provide reliable prototype measurements of thermal-stress patterns around a complete transverse section of a ship. These results are compared with stresses computed by the theory of simple beams under arbitrary temperature distributions across their section. Bridge-type SR-4 strain-gage assemblies were developed for attachment directly to the structure. A variety of temperature conditions were observed and corresponding strain measurements taken. The results are consistent and give a reliable picture of thermal-stress conditions in a ship. They also verify the prediction of thermal stresses afforded by the simple-beam theory.

J. L. Meriam, R. F. Steidel, G. W. Brown and P. T. Lyman are associated with University of California, Berkeley, Calif.

Paper presented at 1959 SESA Spring Meeting held in Washington, D. C., on May 20-22.

Nomenclature

σ_l = longitudinal stress, psi.
σ_t = transverse stress, psi.
ν = Poisson's ratio, 0.29 for steel.

$\bar{\epsilon}_l$ = total longitudinal strain, in./in.

$\bar{\epsilon}_t$ = total transverse strain, in./in.

α = linear coefficient of thermal expansion 6.7×10^{-6} (in./in.)/° F.

E = modulus of elasticity 30×10^6 psi for steel.

T = temperature rise above the arbitrary zero reference temperature, ° F.

A = total cross-sectional area of steel in the transverse section

ΔA = cross-sectional area of each respective element of the transverse section. The ΔA's were chosen, for the most part, so that the boundary between ΔA's lay halfway between adjacent strain gage and thermocouple locations.

$\bar{y}$ = distance from the horizontal centroidal axis (z) of the section to the centroid of the element ΔA involved (plus above and minus below axis).

$\bar{z}$ = distance from the vertical centroidal axis of the section to the centroid of the element ΔA involved (plus to starboard and minus to port).

$\bar{I}_{\bar{z}}$ = moment of inertia of the transverse section about the horizontal centroidal axis of the section.

$\bar{I}_y$ = moment of inertia of the transverse section about the vertical centroidal axis of the section.

Introduction

Up to the present time, relatively little attention has been devoted to the problem of the design of structures subject to stresses caused by thermal gradients. The wide use of welding in ship construction and the record of ship failures during and following World War II have drawn attention to the existence of thermal stresses of appreciable magnitude in ships. A limited number of investigators have worked on this problem, chief among whom are the following:

Hurst[1] in 1943 described a method of calculating thermal stresses based on the assumption of uniaxial stress, no shear, and symmetry port and starboard. Stresses were computed from the difference between the actual temperature distribution and an assumed linear "virtual" temperature distribution. Adequate experimental data for comparison purposes were lacking.

Howe[2] in 1946 measured stress changes in the main decks of a Liberty, Victory and C-2 refrigerated cargo ship. These measurements, however, were quite limited, and no stress distributions corresponding to temperature distributions were determined. Main-deck stress changes of 4800, 6000 and 10,000 psi, respectively, in these three ships were measured corresponding to a diurnal change of about 50° F in the main-deck temperature. Measurements were made with a "bridge" type gage using SR-4 resistance wire strain gages on one side only of the measuring straps.

The Admiralty Ship Welding Committee's tests[3] on the tanker M.V. *Neverita*, reported in 1946, contain a small section on thermal stresses. These were measured during one 12-hr period during which a maximum difference of about 30° F between the deck and bottom temperatures was recorded. Stresses were not measured but were computed by the method of Hurst[1] using the measured temperature distribution. Only 11 temperature stations were used in the transverse test section, so that a complete picture of temperature distribution was absent.

Corlett[4] in 1950 studied the effect of temperature gradients on the stresses in ships constructed with a steel hull and an aluminum superstructure with the aid of models. Temperature gradients were imposed through water immersion and electrical heating. Longitudinal strains around a transverse section were measured. Model stresses obtained from the strain-gage data compared favorably with stresses calculated by considering a one-dimensional state of longitudinal stress as was assumed by Hurst.[1] This investigation did not involve any stress measurements on actual ships.

Jasper[5, 6] in 1955 observed a temperature stress change as high as 10,900 psi on the tanker Esso *Asheville*. Two gage stations in the main deck near the gunwale were employed in these tests. Corresponding changes in deck temperature were not recorded. Furthermore, the gages were mounted under a steel protective box which may have modified the temperature and corresponding stress measurements. Jasper adapted the method described by Timoshenko[7] to the computation of thermal stresses in a box beam from a given temperature distribution. He then applied the method to an assumed temperature distribution in a T-2 tanker and to Corlett's[4] model results as a check.

Hechtman's[8] report on thermal stresses in ships, dated October 1956, contains a lengthy discussion of temperature stresses and includes numerous stress curves based largely on assumed temperature distributions.

The series of tests performed at the University of California had the objective to provide a limited but reliable number of temperature and corresponding stress distributions over a complete transverse section of a ship under measurable and partially controlled conditions. The tests would serve: (1) as a valid base of reference against which the validity of proposed theoretical methods could be checked, (2) as the basis for the design of models for future thermal-stress studies and (3) as a proving ground for additional temperature-stress instrumentation for any future shipboard experiments.

The tests were sponsored by the Society of Naval Architects and Marine Engineers and were carried out by personnel of the Department of Engineering of the University of California, Berkeley. The test vessel, the S.S. *Boulder Victory*, was furnished by the Maritime Administration from its reserve fleet. The S.S. *Boulder Victory* in the test positions is shown in Fig. 1.

Temperature and Thermal Stress Conditions

The relationships between stress, strain and temperature change T for a biaxial state of stress are well known. They are:

$$\sigma_l = \frac{E}{1-\nu^2}(\bar{e}_l + \nu\bar{e}_t) - \frac{\alpha ET}{1-\nu}$$
$$\sigma_t = \frac{E}{1-\nu^2}(\bar{e}_t + \nu e_l) - \frac{\alpha ET}{1-\nu} \qquad (1)$$

The stresses represent elastic changes in stress corresponding to the temperature change T. Each of the total strains represents the sum of the strain (e_l, e_t) associated with the elastic change of stress and the strain (αT) associated with an unrestrained thermal expansion. Thus,

$$\bar{e}_l = e_l + \alpha T, \qquad \bar{e}_t = e_t + \alpha T$$

Substitution of these sums into the foregoing general equations gives

$$\sigma_l = \frac{E}{1-\nu^2}(e_l + \nu e_t)$$
$$\sigma_t = \frac{E}{1-\nu^2}(e_t + \nu e_l) \qquad (2)$$

which are identical with the stress-strain relations for constant-temperature conditions.

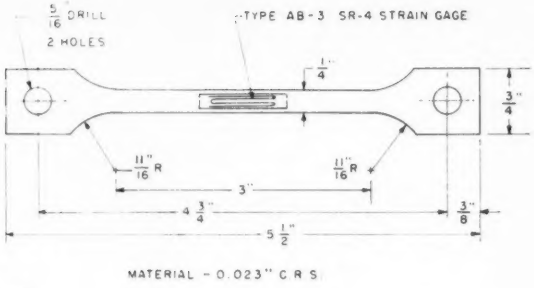
Thus, stresses arising from temperature gradients may be obtained by measuring total strains, eq 1, or merely the elastic part of the total strain, eq 2. The measuring of total strain requires fewer gages than does the measuring of elastic strains. However, the inherent difficulties in using a single constant-temperature dummy gage prompted the decision to use temperature compensatory gages at each gage location and to make computations by eq 2.

Two types of strain measurements were required. First, for all gage locations above the waterline and accessible from both sides of the plate, strains were measured on both surfaces of the plate so that the average of these readings was used to represent the mean strain. A linear strain distribution across the thickness of the plates was assumed since the changes which occurred were elastic. Second, for gage locations below the waterline, where accessibility was limited to one side of the plate, a bridge-type gage was used which depends on an extrapolation of strain readings to get the middle-of-plate strains.

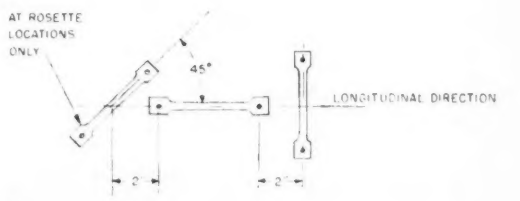
As a direct check upon the validity of a theoretical calculation of thermal stresses, the method cited by Timoshenko[7] and used and discussed by Jasper[5] was employed. The calculations of stress were based upon the actual measured temperatures. This method assumes that the elementary theory of beams is valid with zero stress in the directions transverse to the longitudinal axis of the ship. Further, it assumes no change in temperature conditions in the longitudinal direction and that the resultant force and moment on a complete transverse section are zero. The computed stress in the longitudinal direction is[5,7]

$$\sigma_l = -\alpha ET + \frac{\alpha E}{A}\cdot\int T\cdot dA + \frac{\alpha E}{I_{\bar{z}}}\cdot y\int T\cdot y\cdot dA + \frac{\alpha E}{I_y}\cdot z\int T\cdot z\cdot dA$$

Using small but finite elements of area, this equation



(a)



(b)

Fig. 2—Strain gage (a) and strain-gage layout (b)

becomes

$$\sigma_l = -\alpha ET + \frac{\alpha E}{A}\sum T\cdot\Delta A + \frac{\alpha E}{I_z}\cdot\bar{y}\sum T\cdot\bar{y}\cdot\Delta A + \frac{\alpha E}{I_y}\cdot\bar{z}\sum T\cdot\bar{z}\Delta A \qquad (3)$$

In using eq 3, all of the metal in the transverse section including coamings, bulworks, bilge keels, etc., was used so that the results would be comparable, as much as possible, with the measured stresses. The average temperature over an element ΔA was used in the computations.

Strain Gaging and Instrumentation

Description and Theory of Strain Gages

The strain-measuring gage used in this investigation consisted of a steel strap upon which were mounted two Bakelite-base SR-4 strain gages. This type of gage was used because of the stability of the Bakelite base and the greater ease of mounting the gages in the laboratory over mounting them directly to the surface of the ship plate. Two SR-4 gages were used for each gage strap in order to cancel bending effects of the strap. A drawing of the gage assembly is shown in Fig. 2.

Due to the enlarged end sections of the gage strap, the average strain between the points of attachment of the gage was less than the strain in the narrow section upon which the SR-4 gages were mounted, and thus the sensitivity of the gage assembly was greater than that of the SR-4 gage mounted directly on the plate surface. This increase in sensitivity was taken into account by increasing the gage factor setting on the strain-indicating instrument.

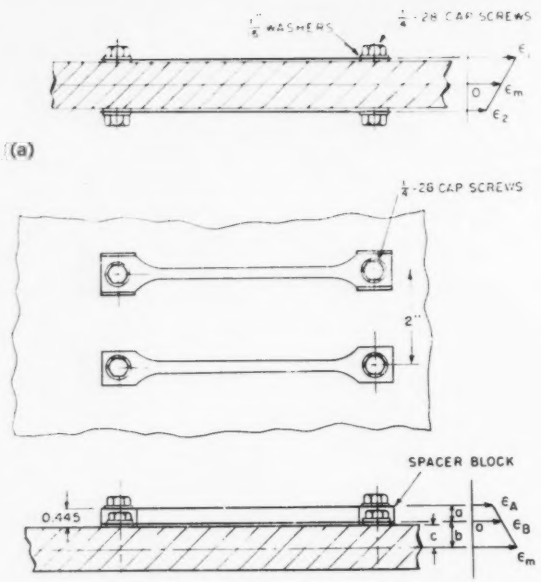


Fig. 3—Strain-gage mountings above water line (a) and below water line (b)

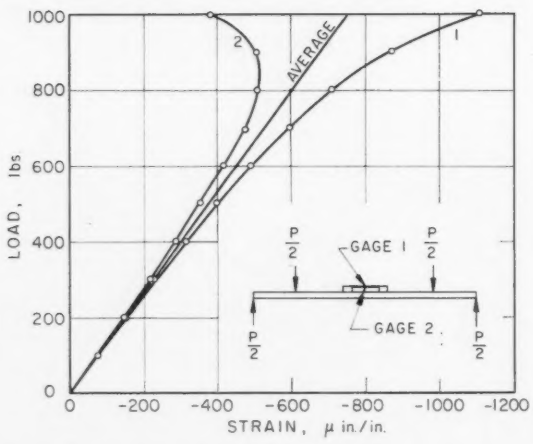


Fig. 4—Strain vs. load for gages or a strap in compression

Two strap gages were used at each location in order to account for local bending effects due to the nonplanar condition of the plates. Above the waterline, gages were mounted on both sides of the ship's plate; below the waterline, the two gages were mounted on the inside surface of the plate at different distances from the plate surface. Figure 3 shows these two configurations. For all locations above the waterline, the mean strain in the plate was found simply by averaging the two strain readings. The strain due to local bending is one-half the difference between the two readings. Below the waterline, it was necessary to extrapolate the reading to the center of the plate. Using the symbols shown in Fig. 3, the extrapolation is accomplished by the formula

$$e_m = K(e_B - e_A) + e_B \quad (4)$$

where $K = b/a$. The strain due to local bending is given by the formula

$$e = \pm \frac{c}{a}(e_B - e_A) \quad (5)$$

Development of Gage

Originally it was intended to mount gages on one side only of the plate at all locations. The assembled gage unit was to consist of two strap gages spot welded at their ends to a spacer block. The spacer block was then to be welded to the inside of the ship's hull. Tests showed that the welding process induced a necessary pretension in both straps. Additional control on the pretension was exercised by means of a mechanical stretcher during welding. However, due to the difficulty in replacing defective gages and the better control on pretension possible by means of controlled heating, it was decided to bolt the gages to the plate at their ends rather than weld them. In addition, it was decided to place the gages on both sides of the plate wherever possible. By drilling completely through the plate, the problem of accurate location was solved, and better accuracy was expected by interpolation rather then by extrapolation.

Manufacture of Gages

Type AB-3 SR-4 strain gages were mounted on both sides of the steel straps following the manufacturer's recommended procedure. The two SR-4 gages were then connected in series and short leads were soldered to the gages. The gages were next carefully cleaned with acetone and coated with a waterproofing compound EC-801 prepared with accelerator EC-807 made by the Minnesota Mining and Manufacturing Co. In order to seal possible pinholes, the gage was given a final coat of a vinyl adhesive, Vyna-Kote, made by the Organic Chemical and Development Co.

Calibration

Gage straps were mounted on a cold-rolled steel bar 1 in. thick, 4 in. wide and 60 in. long. The bar was subjected to both tension and bending tests. These tests were performed both to determine the reliability of bolting the gages in place and also to calibrate them. Fourteen gages were selected at random and attached to the bar in pairs. Two A-5 SR-4 gages were also mounted on the bar for a reference reading. With the gage factor set to the manufacturer's figure supplied with the AB-3 gages, the readings indicated an increase in sensitivity as was expected. An average calibration factor of 0.868 was determined from these 14 gages. The maximum deviation from this figure was 1.12%; the average deviation was 0.62%.

On the bending tests, the strains recorded by the straps and corrected for the increased sensitivity were extrapolated by eq 4 to the center of the bar. The deviation from zero was well within 10 micro-inches per inch in all cases. Calibration curves are included as Fig. 4.

Sixty gage straps were selected at random and loaded with an axial load of 100 lb. The average strain was 600 micro-inches per inch with a maximum deviation of 3% and an average deviation of 1.3%.

Attachment

A template was made to locate the holes to be drilled and tapped at each gage location. The strain-gage configuration is shown in Fig. 2. A drill press with an electromagnetic base was used for drilling and power-tapping all holes. This drill press was easily moved to each location and assured that the tapped holes were perpendicular to the plate. Each location was ground to bare metal to assure a good seat for the gage strap and to make certain that the gage was free of the plate between the points of attachment.

One end of each gage was bolted securely in place by means of a hardened, high-carbon $^1/_4$-28 cap screw. A torque of 150 lb-in. was used to secure the screw. Preliminary tests had indicated that 50-lb-in. torque was sufficient to prevent slipping of the strap up to a load corresponding to the yield strength of the strap. The 150-lb-in. torque was well below that necessary to shear the screw.

Wiring

Lead wires were strung and brought to the central switching panel mounted in an instrument house constructed on the lower 'tween deck. All soldered joints were waterproofed, and the lead wires secured to the ship so that no loose wires were hanging or lying free. All wires outside the ship were placed in pipes and carefully waterproofed wherever they were brought out to the gages. Flat cables of 20-gage, stranded, vinyl-insulated wire made by the Spectra-Strip Co. were used. Each cable consisted of all the wires necessary for one gage location. In this way, dummy and active gage leads were all of the same length and subject to the same temperature variation. A total of 51,000 ft of single-conductor wire was used to install the gages.

Presetting

Since the gage straps could not support compression, it was necessary that they always be in tension. The maximum expected strain was 300 micro-inches per inch. Consequently, the gages were all preset to a nominal tension of 500 micro-inches per inch. Presetting was accomplished by carefully heating the gage strap with a soldering iron and then immediately bolting the free end of the strap using a torque wrench. The amount of heating was determined by observing the strain indicator reading as the heat was applied. Previous tests had been made to determine the relation between indicator reading and temperature of the strap. An average drop of 500 micro-inches per inch due to heating was found sufficient to produce 500 micro-inches per inch residual tensile strain in the strap after bolting and cooling.

Figures 5 and 6 are photographs of completed gage locations, respectively, in the side shell and on the tank top.

Thermal Protection

Simultaneous temperature measurements of the strap and the plate indicated that an appreciable temperature difference existed. This condition was more pronounced at high temperatures with an air stream passing over the gage. Measurements made on the deck of the ship under average weather conditions using an unprotected gage indicated a maximum temperature difference between the deck and the gage strap of $2^1/_2$° F. The temperature of the strap was brought up to that of the plate by coating the strap with a medium-grade cup grease, being certain that the space between the deck and the strap was filled, thereby eliminating convection directly from the strap. All gages in the upper and lower 'tween deck spaces and on the outside of the ship were greased in this way. The temperature difference between strap and plate in the lower hold was negligible. Therefore gages in this space were not greased. Those gages on the outside of the ship were also sprayed with an acrylic lacquer in order to provide a film over the

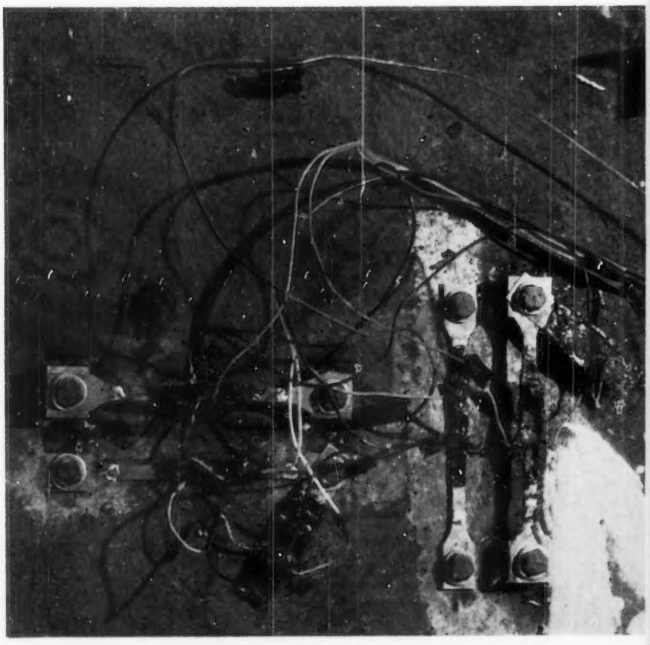


Fig. 5—Strain gages, inside ship, on tank top

grease. The gage area was then repainted with the same primer and boot-topping paint used on the entire test section. The greasing also made it certain that the gage was kept free of any foreign matter that might prevent the unrestricted motion of the gage. Figure 6 is a photograph of one completed exterior gage.

Gage Layout

Longitudinal and transverse gage assemblies were placed at each gage location around the test section. Figures 7 and 8 show the layout of the gages. At gage positions 3-1, 4-1, 5-3 and 6-3, an additional gage was placed at 45 deg to the ship's axis to form a rosette. This gage was used to determine the direction of the principal stress at these four locations. Gages were also placed along longitudinal sections extending 12 ft forward and aft of the transverse section at positions 1-9, 2-9, 3-5 and 4-5.

Tank Gages

Three gage locations were situated on the tank bottoms of the number 3 hold. Since these tanks were to be ballasted with fresh water, special precautions were taken to insure that the gages would be waterproof. The gages were completely encased in a water barrier, and the leads were brought out through a length of copper tubing. The copper tubing passed through a packing gland in the tank top. A sketch of one installation is shown in Fig. 9.

Switch Panel

A switch panel was designed to permit the simultaneous measurement of three strains and two

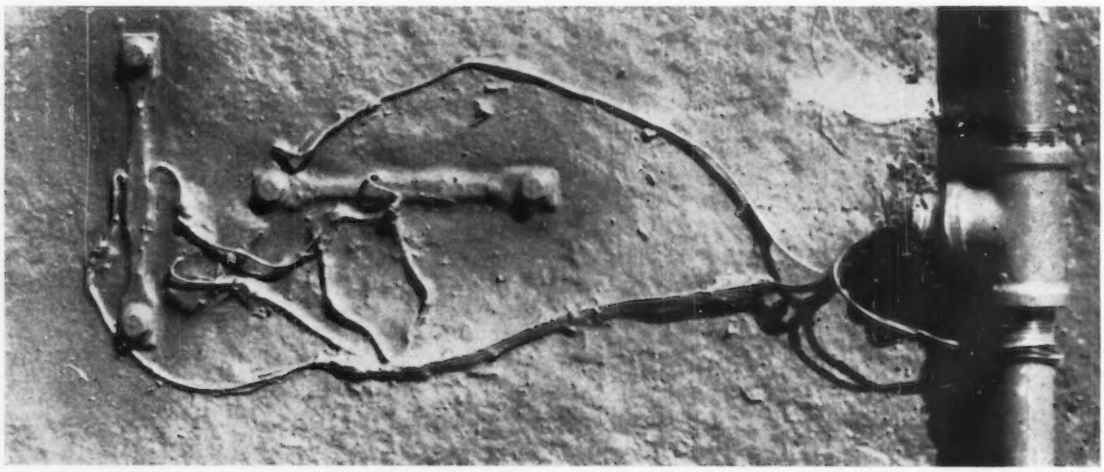


Fig. 6—Strain gages, exterior

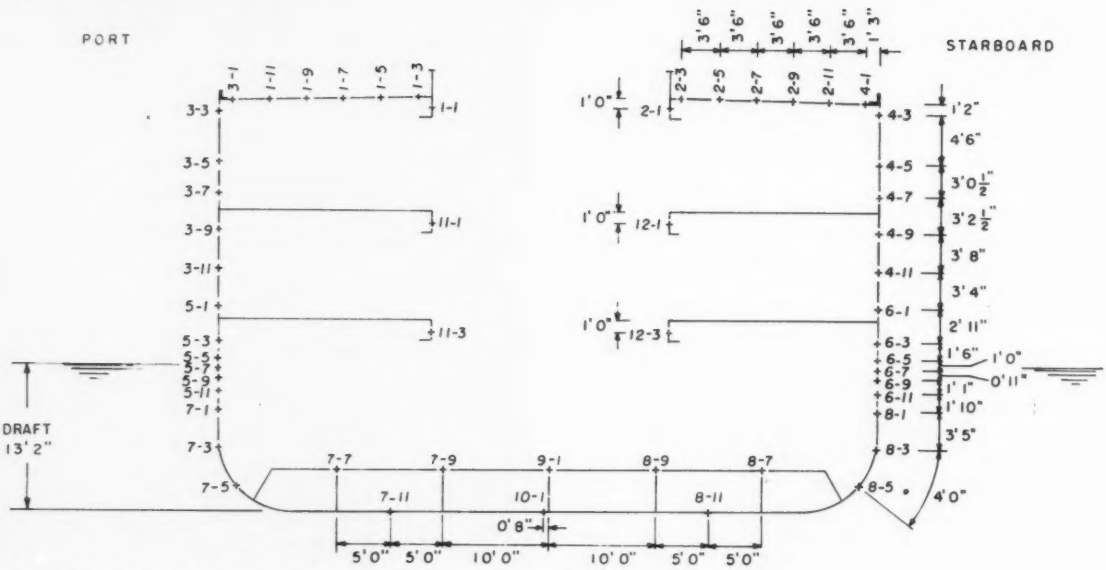


Fig. 7—Gage location and numbering for the transverse test section

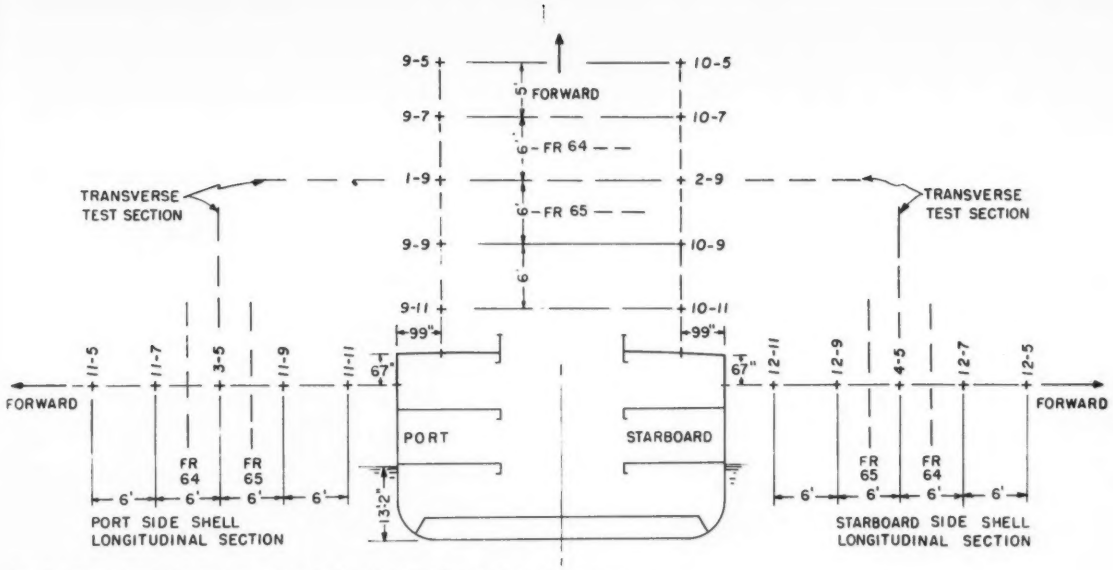


Fig. 8—Gage location and numbering for the longitudinal test sections

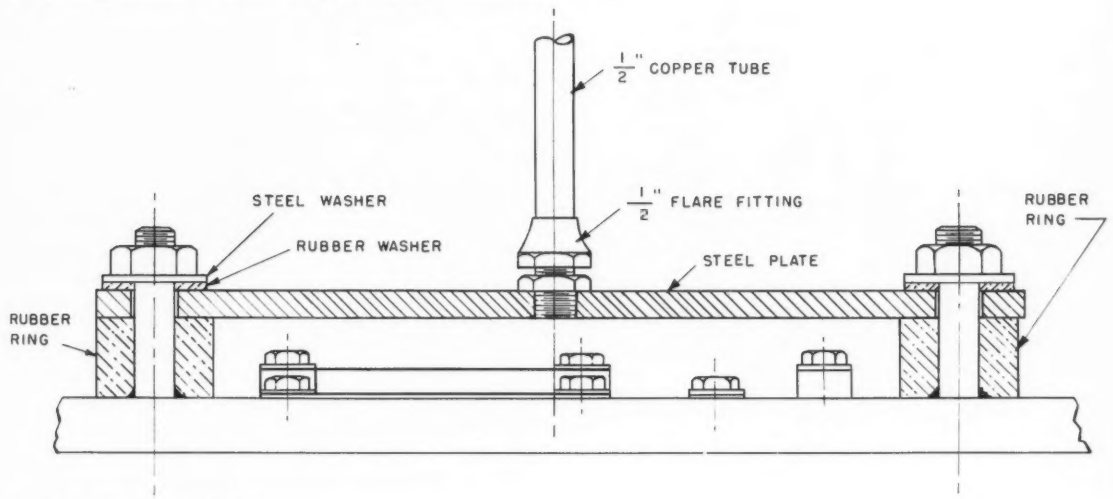


Fig. 9—Tank-gage installation

temperatures at each of 144 gage locations. The panel was equipped with Leeds and Northrup 31-3 series rotary selector switches. These switches employ low-resistance, 0.001-ohm, silver-silver alloy contacts. Life tests of these switches over a million cycles show a change of contact resistance of less than 0.0005 ohm, which corresponds to a change in strain of about 1 micro-inch per inch for a 240-ohm strain gage. The change of contact resistance has no effect on the temperature measurements. The switching panel was so arranged that it required only one person to take all necessary strain and temperature data.

A schematic wiring diagram of the strain-gage switching is shown in Fig. 10. The operation of only three switches permitted the selection of any one strain gage out of a possible 432 gages. The operation of the switching panel was as follows: (1) The 12-pole "switch-selector switch" was used to select any one of the 12 "location-selector switches." The 12-pole "location-selector switch" was used in turn to select one of its 12 gage locations. (3) The 3-pole "gage-selector switch" was used to select from the gage location one of the three following gages: longitudinal, transverse or the 45-deg leg. The similar operation of only three switches permitted the selection of any one thermocouple out of a possible 288 thermocouples.

Strain Recording

Individual strains were measured on a Fairchild Model 101B strain indicator. The indicator employs a servo-balanced bridge circuit to measure the change of resistance of the strain gages. A Coleman shaft position analog-to-digital converter

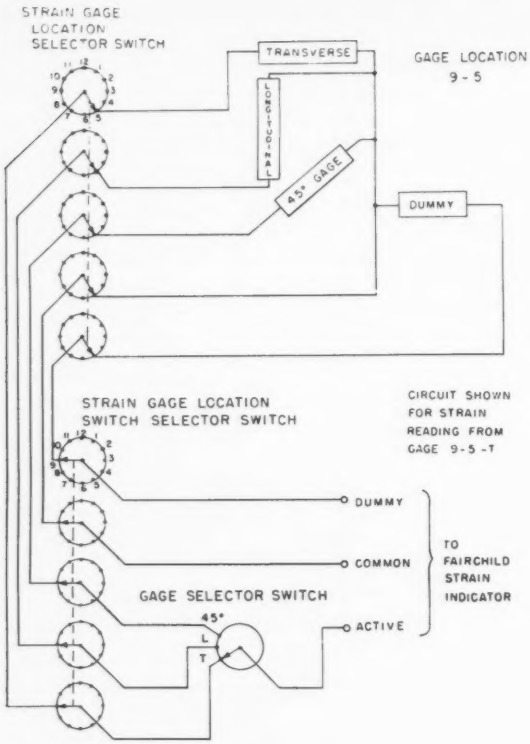


Fig. 10—Strain-gage wiring

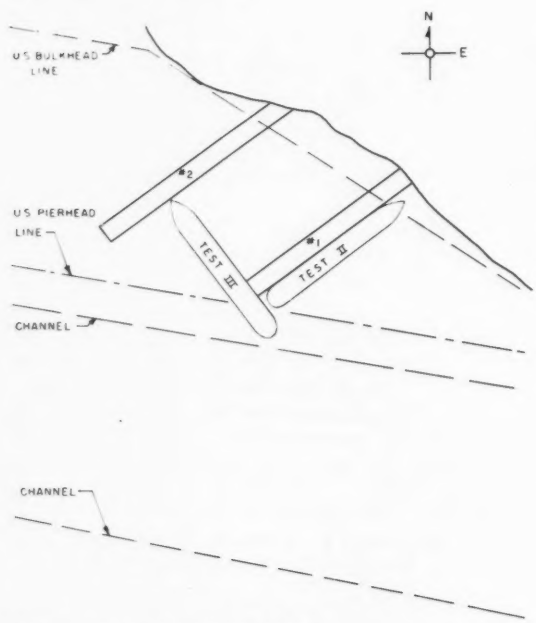


Fig. 11—Position of ship during tests

was mounted in the Fairchild indicator to permit the printing of digital strain data.

All strain and temperature data were recorded with a Clary numerical data printer. The data

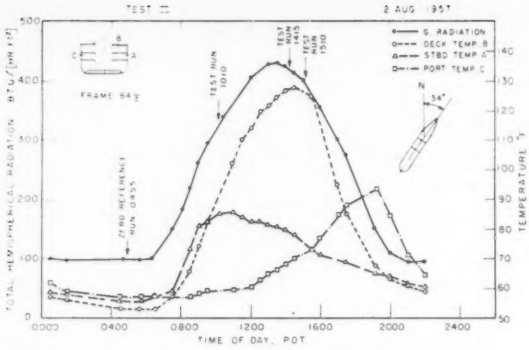


Fig. 12—Temperature and radiation control plot—Test II

printer was designed to print simultaneously a three-digit identification code, the temperature and the strain. The complete data-printing system consisted of the digital converters, the data printer and a programming unit. The functions of the Moseley programming unit were to act as the system power supply, to energize the converters, to mute the servo systems of the indicators during printing and to actuate the printer. The system was actuated by the use of a "read out" push button mounted on the switching panel.

Test Procedure and Results

Following the completion of the instrumentation and the necessary check on the operation of gages, thermocouples and instruments, a trial test (Test I) was conducted. In addition to a check on the response of all instruments, the test showed that a significant range of temperature conditions, including symmetrical and asymmetrical ones, could be observed with the ship in its fixed position alongside the dock. On Aug. 2, 1957, with the ship in position II shown in Fig. 11, a full test was accomplished. The temperatures at three different positions A, B and C at the test section, Fig. 12, and the total hemispherical radiation were monitored throughout the day.

Initial readings for strains, temperatures and deflections were taken at midnight on August 1 and again at 0455 on August 2 to establish the zero reference conditions for minimum temperatures. Monitoring the variation of side shell and deck temperatures and solar radiation versus time, it was possible to anticipate the most significant conditions of temperature variation and to take complete sets of data at these times. Complete sets of strain, temperature and deflection data were taken when the difference in temperature rise between the two side shells was a maximum 1010, when the intensity of solar radiation was nearly a maximum 1415 and when the deck reached its greatest temperature 1510. From 15 to 20 min were required to take a complete set of strain-temperature data. Measurements of air tempera-

ture and humidity, wind velocity and direction, current, water temperature and air temperatures inside the hold spaces were made during the day. Additionally, a record was kept of the sun's altitude and bearing at the time of tests and of the position of shadows in the test section.

Changes in stress figured from the condition at 0455 which was taken as the zero reference were computed from the measured elastic strain increments for each of the three observation times. The longitudinal component of these stresses for one condition, namely that at 1415, are shown in Fig. 13.

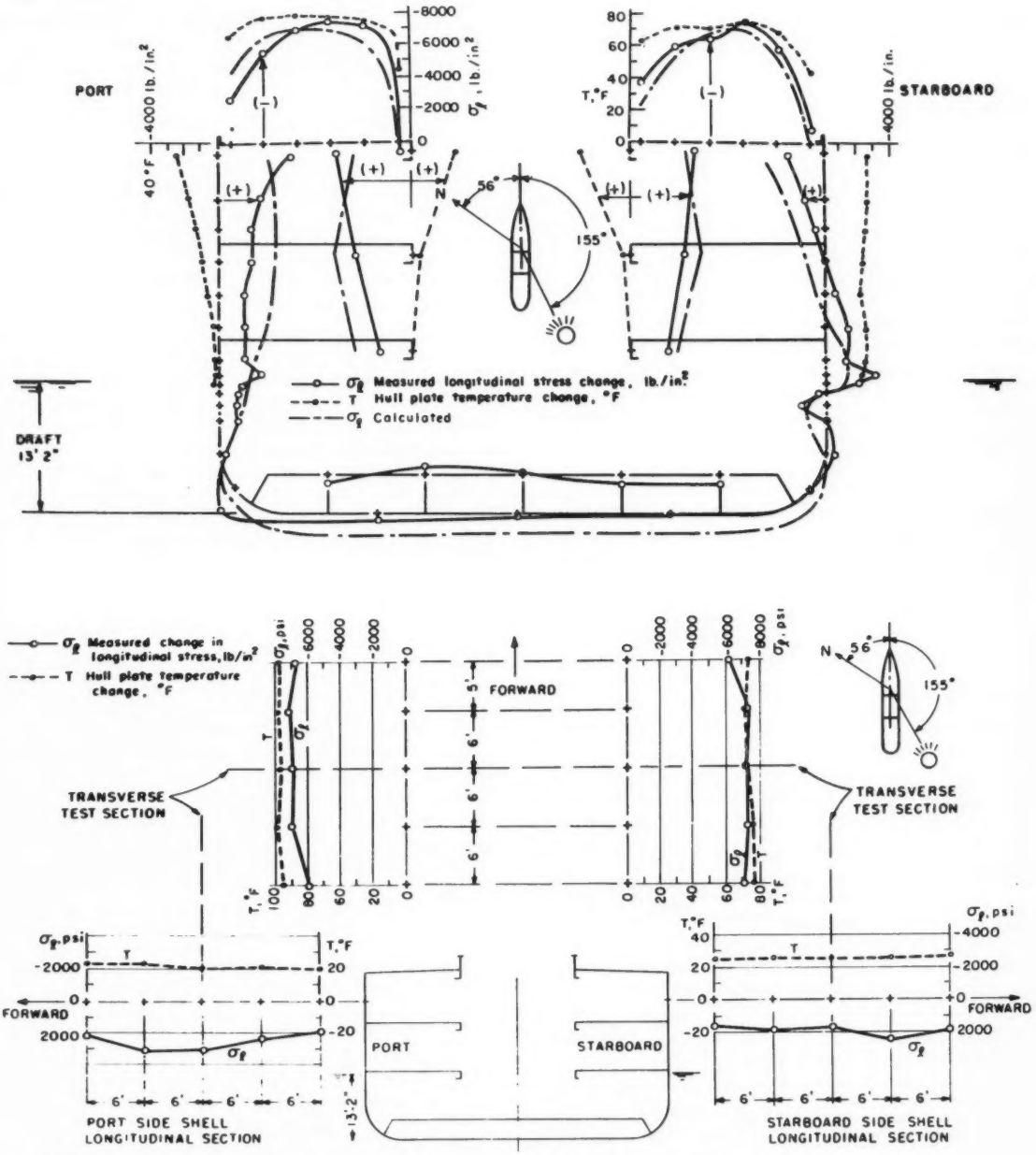


Fig. 13—Temperatures and corresponding thermal stresses on the transverse and longitudinal test sections. Temperatures and stresses represent changes from a zero reference datum at 0455 PDT, 2 Aug. '57. (+ Tension, — Compression)

Test II Time: 1415 PDT Ship's heading: 56 deg true Sun's altitude: 68 deg
Date: 2 Aug. '57 Sky: Clear Sun's azimuth: 99 deg

The predicted longitudinal stresses based on eq (3) are shown in this same figure as the dashed line. The distribution of temperature change from the zero reference condition is shown as the dotted line.

Following the completion of Test II, the data were processed to obtain the stress changes corresponding to the measured temperature changes. Other tests were made but Test II is exemplary.

A complete account of the results of the other test conditions observed may be secured through the Society of Naval Architects and Marine Engineers.

Discussion

Agreement Between Theory and Experiment

The most striking observation apparent from the stress plots, Fig. 13, is the excellent agreement between the stresses computed from the measured strains and the stresses computed from eq 3 using measured temperatures. At the conditions of maximum deck stress in Test II, the measured peak deck stresses were from 5 to 7% higher than the calculated values.

This correlation exists in the neighborhood of the test section where transverse stiffness due to bulkheads is a minimum and where gradients of strain and temperature in the longitudinal direction are small.

A comparison between the curves of deck temperature versus time and the curve of hemispherical solar radiation versus time, Fig. 12, shows a very close correlation in both shape and time response. It is noted that the peak of deck temperature was reached only one hour after the peak of solar radiation, so that the thermal response of the ship structure to radiation is remarkably rapid.

Adequacy of Instrumentation

One of the important objectives of the investigation was the development of adequate instrumentation for the measurement of thermal stresses. The instrumentation on the *Boulder Victory* proved to be adequate for this purpose. In making this statement the following specific points should be emphasized:

1. The effect of shadows on the ship's plating was seen to be extremely local, so that any strain or temperature-measuring instruments placed on the exterior surface should be as close to the hull as possible to minimize shadow effects.

2. Strain gages attached to exterior surfaces should be connected to the steel plate by as short a thermal path as possible. This connection should provide for a maximum of conduction and should eliminate convection heat loss for the gage. The technique used in the present study of filling the space between the gage and the plate with grease satisfied these thermal requirements. The stability of this treatment over a period of, say, six months or more is not certain and would have to be investigated.

3. The heat-transfer characteristics of the exterior surface of the strain gage and those of the plate should be as near the same as possible. Using the same paint for the exposed gage-strap surface as was used for the deck and side shell fulfilled this requirement.

4. The technique developed for pretensioning the straps gave no trouble and assured a consistent response of the gage straps within their linear stress-strain range. Caution would have to be exercised if similar gages were used under circumstances where the strains would exceed the elastic limit. By keeping a record of the initial pretension readings, however, a strain which approached or exceeded the elastic limit could be detected.

5. The two major advantages of automatic balancing and recording of all strain and temperature readings are considered important to the success of the tests. These are the rapidity with which the entire transverse section of the ship could be scanned (about 15 min) and the virtual elimination of human error when such a large number of individual readings are involved.

6. As was mentioned at the outset, it was contemplated that a constant-temperature compensating gage could be used as a reference leg of the bridge if the temperatures of both compensating and measuring gages were accurately known. It was discovered that the sensitivity of the gages to temperature variations was so great that it was necessary to use individual compensating gages at each gage location on each side of the plate.

Conclusions

1. The theoretical solution for longitudinal thermal stresses in a ship's hull girder by eq 3 is in excellent agreement with the actual measured stresses around a complete transverse section at hatch No. 3. This conclusion applies for temperature distributions which are both symmetrical and asymmetrical with the vertical center line of the section.

2. The strap gage and associated instrumentation used for the *Boulder Victory* tests proved entirely adequate for the measurement of thermal stresses.

Bibliography

1. *Hurst, O., "Deflections of Girders and Ship Structures: A Note on Temperature Effects,"* Trans. Inst. Naval Arch., 85, *74 (1943).*
2. *Howe, E. D., Boodberg, A., and O'Brien, M. P., "History of Residual Stresses in Welded Ships (NS-305), Part III—Temperature Studies of Liberty, Victory, and Refrigerated Cargo Ships," OSRD No. 6590, Feb. 25, 1946.*
3. *"Hogging and Sagging Tests, M.V. Neverita," Admiralty Ship Welding Committee, 1946.*
4. *Corlett, E. C. B., "Thermal Expansion Effects in Composite Ships,"* Trans. Inst. Naval Arch., 92, *376 (1950).*
5. *Jasper, N. H., "Temperature-Induced Stresses in Beams and Ships," T.M.B. Report 937, June 1955.*
6. *Jasper, N. H., "Service Stresses and Motions of the Esso Ashville . . .," T.M.B. Report 960, September 1955.*
7. *Timoshenko, S., and Goodier, J. N., "Theory of Elasticity," McGraw-Hill, p. 401 (1951).*
8. *Hechtman, R. A., "Thermal Stresses in Ships," Oct. 30, 1956; a report presented to the Ship Structure Committee (c/o U. S. Coast Guard).*
9. *Leszynski, S. W., "Strain Gage Applicator,"* Proc. SESA, 13 *(1), 161 (1955).*

sesa news

Philadelphia Spring Meeting to Feature Seminar on Computers

"The Role of Computers as a Tool in Experimental Mechanics" is the title of an evening seminar to be held during the 1961 Spring Meeting of the SESA at the Benjamin Franklin Hotel in Philadelphia, Pa., on May 10–12. Chairman of the committee arranging details for the seminar is Professor James Stone of Drexel Institute. Plans are being made for an additional plant tour to study the digital computer installation at the University of Pennsylvania, and the analog and digital equipment at Drexel. The added features at the annual Spring Meeting have been announced by Stewart W. Sennet and Louis B. Moffet III, co-chairmen of the affair for which the Delaware Valley Chapter is acting as host.

Technical Papers Sessions

The main event of the three-day program will be the presentation in five sessions of 15 to 20 papers dealing with various instrumental methods of measurement and analysis in the field of experimental mechanics. The five sessions are divided into two main groupings. One group of four sessions will consist of the presentation of papers selected by the Papers and Publications Committee, under the chairmanship of Roscoe Guernsey, Jr. These selections will be published in EXPERIMENTAL MECHANICS and will also appear subsequently in the bound volumes of SESA *Proceedings*. The other group will consist of papers on the meeting theme "Experimental Mechanics at High Temperatures." Papers for this special session will be accepted for consideration up to April 15th by the Technical Sessions Committee for the Spring Meeting, and should be sent to Mr. Sennet at the Mechanical Research Laboratory, E. I. du Pont de Nemours Co., Inc., 101 Beech St., Wilmington, Del. Presentations made at the theme session may be considered for later publication in the *Journal* and the *Proceedings* provided they are submitted in due course to the Papers and Publications Committee for review.

Other Events

Carrying out the theme of the meeting, a plant tour has been arranged to the well known Eddystone Power Station. Here the visitors will be able to inspect the high-temperature high-pressure equipment employed by this installation for power generation. Some first-hand knowledge can be gained of the many problems surmounted in the design and construction of this high-pressure system.

Midway in the program an evening banquet has been scheduled to afford a few hours relaxation and fellowship during the busy schedule. Frank G. Tatnall, the first man to be elected to Honorary Membership in SESA, will act as toastmaster. Arrangements are also under way for securing an outstanding nationally known personality to address the group on a timely subject.

Further details on the Spring Meeting will follow in later issues of EXPERIMENTAL MECHANICS.

International Congress to Include Largest Exposition

Plans for the First International Congress on Experimental Mechanics, to be held in New York City, Nov. 1–3, 1961, include sufficient floor area to make the Exposition the largest on record for a SESA meeting. The entire mezzanine floor of the Hotel New Yorker will be given over to Society activities during the Annual Meeting for which the New York Metropolitan Section will serve as host. This arrangement will allow ample room for registration and meeting area while reserving the entire balance of the floor for exhibition purposes. Having technical activities, Society business and the exhibits together on one level will help to stimulate booth visitations.

The unusual opportunities offered by an international meeting held in New York City is expected to attract a large number of exhibitors. For the first time, foreign manufacturers are invited to apply for space. As in previous years, a number of educational exhibits are expected to show typical examples of the latest work being carried on in governmental, private and university research laboratories.

The subject for an appropriate theme for the Congress has been discussed and technical papers are being solicited to carry forth the idea. While the exact title will be announced later, the theme will be concerned with the role being played by experimental mechanics in providing a critical re-evaluation as well as new perspectives on fundamental theory. A definite move in this direction is believed necessary by many engineers, scientists and educators. Present-day demands on the performance of materials and structures have brought into sharp focus an urgent need for re-defining theories and developing new ones.

One of the main features of the International Congress will be the presentation of papers by noted scientists and engineers from abroad. Arrangements are being made for representatives from France, Germany, Italy, Netherlands, Portugal, Spain, Sweden and the United Kingdom. Although some commitments have been received, a detailed announcement will be published when the program is more nearly complete.

All papers intended for presentation at this meeting should be submitted before May 31, 1961, and should be addressed to Dr. Roscoe Guernsey, Jr., Chairman, Publications Committee, Society for Experimental Stress Analysis, General Engineering Laboratory, General Electric Co., Schenectady 5, N. Y.

Each year an outstanding engineer or scientist is selected to present the traditional Murray Lecture, the highest

Employment Service Bulletin . . .
. . . a Service Feature for Members

As a service to SESA membership—both corporate and individual—a column will regularly carry information regarding "Situations Wanted" and "Positions Available." Members are encouraged to send a brief of particulars to Experimental Mechanics, 21 Bridge Square, Westport, Conn. An insertion will appear once and will be limited to 75 words. No charge will be made for this service.

honor SESA can award. So honored this year is Professor Egon Orowan of Massachusetts Institute of Technology.

More details on this will appear in a later issue of EXPERIMENTAL MECHANICS.

International Symposium on Photoelasticity Scheduled for Chicago

On October 29th to 31st, the International Symposium on Photoelasticity will be held at the Illinois Institute of Technology, Chicago, Ill. One of two international conferences in fields related to experimental mechanics, the Chicago symposium immediately precedes the International Congress on Experimental Mechanics being held in New York City on November 1st to 3rd.

The proximity of the two meeting dates will permit those wishing to attend both conferences to do so in the shortest time possible. This arrangement should benefit especially those visiting this country from abroad who may wish to keep their stop-over at a minimum.

The deadline for submission of manuscripts for the Chicago meeting was Jan. 31, 1961. However, those seeking further information should contact Paul D. Flynn, Sec., International Symposium on Photoelasticity, Illinois Institute of Technology, Chicago 16, Ill.

Further detailed information on the New York International Congress will be found elsewhere in this issue.

Reports on European Permanent Committee for Stress Analysis

In a letter written to Past-president B. J. Lazan and dated Nov. 23, 1960, A. F. C. Brown, chairman of the Joint British Committee for Stress Analysis, had the following interesting information to relate:

"I was glad to see that you had been elected President of the SESA, and send you my hearty good wishes for a successful period of office. Over here we watch the development of the Society with great interest. Bound volumes of the *Proceedings* are to be seen on the desks of a surprisingly wide range of scientists and engineers, and we welcome the possibility of earlier publications through the medium of the new monthly *Journal of Experimental Mechanics*.

"I thought you might be interested to hear of a recent development on this side of the Atlantic in the field of stress analysis. The Stress Analysis Group of the Institute of Physics (now the Institue of Physics and the Physical Society) was formed some 14 years ago and its first chairman was Colonel H. T. Jessop, whom I believe you have invited to attend the SESA International Conferences next year. Our little Group has been holding annual conferences rather on the lines of those held by the SESA, but on a much smaller scale. In 1959 the annual conference was held at Delft at the invitation of the Dutch TNO. This conference was made open to anyone interested; representatives from 16 countries attended and it was voted an entire success. Since then a Permanent Committee for Stress Analysis, open to all countries in Western Europe has been set up to arrange for similar conferences periodically with different countries acting as host in turn. The conferences will be open to all nationalities and will probably be held at four-yearly intervals in between the International Congresses for Applied Mechanics.

"The British Joint Committee for Stress Analysis was formed to appoint British representatives to serve on the European Permanent Committee. It now seems, however, that other more general functions in relation to stress analysis might form part of the interest of this committee. For example it has been decided to keep a register of names and addresses for mailing items of stress analysis news. The SESA is already on this list.

"As I expect you know, it is intended that a conference on stress analysis shall be held in Paris in 1962. The organization will be entirely in the hands of a French committee and it is in fact being administered through the offices of GAMAC. An early decision of the Permanent Committee was that, once the location of the next conference was settled, the host country would carry the entire responsibility for its organization."

CORRECTION:

On Saint Venant's Principle Under Dynamic Conditions

Discussion by A. J. Durelli and J. W. Dally

We regret that the three photographs appearing on page 20 of the January issue as Fig. D2 -- (a), (b), (c) are not in proper vertical order. To get correct position, place the top photo at the bottom and then raise all three. After arranging, turn top photo 180 degrees. Of course, all captions and identifying marks remain unchanged.

The Editors

A SALUTE TO CHARLES V. MAHLMANN

It is well known that our Society has been operated since its inception largely by the volunteer efforts of many devoted members. Not so well known, perhaps, is the valuable contribution made by C. V. Mahlmann who, for the past decade, has been managing editor of the *Proceedings*. For this reason and because he is now yielding this post to a full-time staff, a salute to his excellent work seems most appropriate.

Few, except those who have been close to the detailed operations of the Society, can fully appreciate the many hours of patient labor that Editor Mahlmann has contributed to the *Proceedings*. He has not only directed production but, in addition, has personally prepared many of the drawings, tables and diagrams as well as the master sheets required for a clear and attractive presentation. His work has supplemented the contributions of the numerous authors and friends of SESA whose efforts have brought international recognition to this publication

In our retiring editor, the Society has found not only an enthusiastic and devoted friend but also an unusual and brilliant man. Charles received his early education in Japan where he learned to speak the language. Entering the University of Dayton (Ohio) at the tender age of 14, he received his M.E. degree four years later. He pursued his graduate studies at several universities—Cincinnati, George Washington, Wisconsin— and earned his master's degree at M.I.T. While at college, "Chuck" gained experience in various aspects of the printing industry—an interest he holds to this day.

In his professional life as an engineer, he has been associated with a number of prominent firms including Emerson Radio, General Motors Corporation and the Instrumentation Laboratory of M.I.T. More recently, he has established himself as a consultant. His unusual capabilities in machine design have earned him an outstanding reputation, especially in the field of electro-mechanical devices. Lately, he has been devoting some time to the design of plastics and to the development of mechanical equipment for the medical profession.

The Society has indeed been fortunate in having a generous and loyal friend with exceptional professional skill and a useful knowledge of the printing industry. To "Chuck" Mahlmann the SESA extends its thanks for a job well done and takes this occasion to wish him continued success.

A Short Intensive Course

on

STRAIN GAGE TECHNIQUES

(with Supplementary Laboratory)

will be given at the

SOUTHWEST RESEARCH INSTITUTE

in

SAN ANTONIO, TEXAS

APRIL 17-21, 1961

Under the Joint Sponsorship of the Southwest Research Institute and the Society for Experimental Stress Analysis

---

For further information write to

Dr. M. M. Lemcoe
Southwest Research Institute
8500 Culebra Road
San Antonio 6, Texas

For details, circle No. 7 on Reader Information Card

NEW SESA MEMBERS

CALIFORNIA

Abrahamsen, Egil
Berkeley

Beidler, Richard
Manhattan Beach

Boukamp, J. G.
Berkeley

Brush, Don O.
Palo Alto

Chin, George
Citrus Heights

Christensen, R. H.
Rolling Hills Estates

Daub, James E.
San Diego

Dow, R. H.
Sacramento

Farnham, Robert B.
Orangevale

La Fleur, George N.
Santa Fe Springs

Luchter, Daniel
Walnut Creek

Lundberg, Anders W.
San Francisco

Makerov, C. E.
Livermore

Meissner, Fredrick H.
Sacramento

Nelson, William E.
Berkeley

Rogers, Floyd E.
Sacramento

Perless, Leonard C.
N. Sacramento

Poucher, Rudy
Sacramento

Pranger, Richard W.
San Diego

Rhoades, Dewey J.
Woodland Hills

San Miguel, Anthony
Pasadena

Sanders, D. E.
Sacramento

Sanders, Robert H.
Sacramento

Sando, M. Dean
Whittier

Steele, R. K.
Carmichael

Takizawa, Megumi
Pasadena

Talmo, Robert E.
Pasadena

Wardein, Harold E.
South Pasadena

CONNECTICUT

McCarthy, Roger J.
E. Hartford

Rossi, Boniface E.
Georgetown

DISTRICT OF COLUMBIA

Johnson, Charles K.
Washington

FLORIDA

Moran, John Patrick
Orlando

ILLINOIS

Kling, Gary H.
Peoria

Klinke, Warren R.
Riverside

Knisely, C. W.
Tuscola

INDIANA

Shang, Jeng C.
East Chicago

Willis, David L.
Greenfield

LOUISIANA

Goodrich, Hale C.
Shreveport

MARYLAND

Abeln, Clarence H. R.
Hagerstown

Browning, Stanley C.
Cumberland

Goeller, Jaques
Baltimore

Liebowitz, Harold
Chevy Chase

Richardson, William B.
Silver Spring

MASSACHUSETTS

Ajemian, J. Charles
Boston

Barry, John D., Jr.
Lexington

Dorsey, James
Winchester

Lemay, Edward J.
Watertown

Ligenza, S. J.
Southwick

MICHIGAN

Ankofski, Donald J.
Hazel Park

Cuff, George E.
Detroit

Nicholas, Grady B.
Milford

Schneider, Lowell Clark
Pontiac

MISSOURI

Davidson, Robert Frederick
Rolla

Thompson, Mason B.
Kansas City

NEW JERSEY

McCall, John T.
Wharton

Norris, David P.
Somerville

NEW YORK

Gopalakrishnan, M. B.
Troy

Kirschner, Francis
Long Island City

Lightstone, John B.
Syracuse

Mains, Robert M.
Schenectady

Marich, Frederick
New York

Miklofsky, Haaren A.
Troy

Pezold, William H.
New Hyde Park

Supensky, Alwin C.
Syracuse

NORTH CAROLINA

Myers, Guy Arnold, Jr.
Raleigh

OHIO

Newman, Robert M.
Cleveland

OKLAHOMA

Vogt, Calvin O.
Tulsa

PENNSYLVANIA

Eng, Gim H.
Pittsburgh

Green, Sidney
Philadelphia

Hawk, Raymond A., Jr.
Reading

Lohmeier, Alfred
Media

Ruddy, Bernard C.
Rydal

RHODE ISLAND

Carlson, Robert Goran
East Providence

TEXAS

Lindgren, D. E.
Houston

Nordin, Obert L.
Houston

Sowers, George M.
Houston

Ursell, C. R.
San Antonio

WASHINGTON

Clarke, William F.
Bellevue

Oakley, Allen G.
Bellevue

Ryan, James A.
Seattle

WISCONSIN

Bier, Donald E.
Milwaukee

Lis, Gordon J.
Milwaukee

CANADA

Duquette, Douglas D.
Windsor, Ontario

Howell, Rev. A. R. CSB
Windsor, Ontario

Takahashi, Saburo Sam
Toronto, Ontario

Toronchik, Jack
Toronto, Ontario

Turner, John Brian
Peterborough, Ontario

OTHER NATIONS

Amstutz, Prof. E.
Zurich, Switzerland

Beguin, Dr. Gilbert H.
Geneva, Switzerland

Dunne, Padraic C.
Extado de Sao Paulo, Brazil

Rivera, Raymundo
Monterey, N. L., Mexico

Sanchez, Roberto
Mexico 12, D. F., Mexico

Santiago, Corro C.
Mexico, D. F., Mexico

Shimada, Heihachi
Sendai, Japan

Quarter Ending on December 31, 1960

Members	**2059**
Honorary Members	**3**
Corporate Members	**85**
TOTAL MEMBERSHIP	**2147**

section reports

Chicago

Semiconductors and Strain Gages

The November 17, 1960, meeting of the Chicago Section was held at Crossley Associates, and the speaker for the evening was James Dorsey, chief engineer, Strain Gage Division of Baldwin-Lima-Hamilton. Mr. Dorsey is a graduate of Massachusetts Institute of Technology. The title of his talk was "I.S.A. Strain Gage Symposium—A Review."

The topic was concerned mainly with semiconductors and the new Baldwin-Lima-Hamilton universally temperature-compensated strain gage. The gage is designed with a platinum-wire compensating element contained within the boundaries of the strain-sensing foil grid. This configuration permits accurate compensation in applications, where rapidly changing temperature high thermal gradients, mechanical conditions, or space limitations make it impractical to mount a conventional "dummy" gage.

Mr. Dorsey related that Baldwin-Lima-Hamilton is keenly interested in the development, characteristics and applications of semiconductors. He reviewed several papers from the recent I.S.A. meeting and presented various data in the form of graphs and charts which showed how the gage factor varied with temperature, strain, impurities, etc. Semiconductors should have applications in the transducer field due to their high gage factor.

Prior to the business meeting there was a dinner at Chef Alberto's.

(Reported by J. C. Tamburino for the Chicago Section)

Hudson-Mohawk

Photo-thermal Elasticity

"Photo-thermal Elasticity" was the topic for the November meeting of the Hudson-Mohawk Section of the SESA at Poughkeepsie, N. Y. Before an audience of approximately 35, George Girard presented a chronological review of the development of the techniques along with a description of experiments currently being conducted by himself and his colleagues at New York University.

This technique, which is, of course, the determination of thermal stresses by photoelasticity methods, uses a birefringent plastic model which not only makes the fringe patterns possible but, fortunately for transient problems, slows the propagation of thermal-shock waves because of its low thermal conductivity. Thermal excursions are provided by dry ice for thermal-shock conditions in a dry ice, alcohol mixture for steady state in order to avoid specimen creep associated with high temperatures. Hysol was found to be an acceptable model material because of its thermal and stress-optic properties, as well as its ease of machining and cementing. Validation of the technique was accomplished by experiments with H-beams and hollow cylinders by comparing photoelastically determined thermo stresses with those computed from imbedded thermocouple measurements.

Dr. Girard also discussed the following experiments recently conducted or now in progress using this technique:

1. The determination of end effects of beams of various aspect ratios with parabolic temperature distribution.

2. Determination of end effect of trimetallic strips.

3. A study of polariscopes integral with the specimen for three dim-

SESA Section Meeting Calendar

February 1

DELAWARE VALLEY Chapter. Steam Turbines Plant, Westinghouse Electric Corporation, Essington, Pa. Technical program on "Measurement of Stresses and Vibration in Turbine Blades During Operation."

WASHINGTON AREA Section. Dinner: 6:15 P.M. Carmack's Restaurant, 4469 Conneticut Ave., N. W. For meeting place call Dr. R. G. Barclay, Emerson 2-8000, ext. 7561. "Fluid Dynamics Research Here and Abroad," by Marshall P. Tulin, Hydronautics, Inc. Free refreshments.

February 2

HUDSON-MOHAWK Section. 8:00 P.M. Latham, Mass. "Thermal Shocks in Solids—Its Generation and Measurement," by F. Wendt, Missile & Space Vehicle Dept., General Electric Co.

February 10

SOUTHWEST Section. Spring Meeting. Admiral Kidd Club, San Diego, Calif. Talk and film on "High-temperature Testing of the X-15 Aircraft," by O. L. Gillette of North American Aviation.

EDITOR'S NOTE: Notices to be published in any one issue must reach SESA headquarters prior to the 15th of the second previous month. Give full information concerning date, time, place, topic and speaker for each meeting.

GIVES TALK ON STRAIN GAGES

James Dorsey is shown addressing the November 17, 1960, meeting of the Chicago Section. He spoke on semiconductors and temperature-compensated strain gages

ensional studies. The cementing of the polariscope into the specimen was found to introduce only small errors.

(Reported by J. P. Partell for the Hudson-Mohawk Section)

New England

Strain Gaging Ships and Submarines

The December 7, 1960, meeting of the New England Section was held at the MIT Faculty Club, Cambridge, Mass. Ralph B. Allnutt, of the David Taylor Model Basin, Washington, D. C., presented a talk entitled "Developments in Experimental Stress Analysis as Applied to Ships and Submarines."

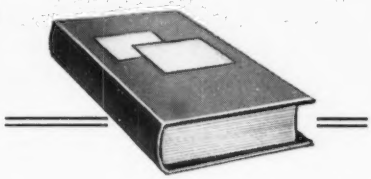
Mr. Allnutt gave an excellent presentation with the aid of slides. He explained that the David Taylor Model Basin was responsible for the experimental development of ships and submarines. It must develop load limits and design data for hull structure. It must find the static and dynamic strength of ships and subs for both peacetime and wartime environment.

The speaker also explained the use and application of strain gages from hulls of deeply submerged submarines to foils on swift (75 knots) hydroplane boats.

He stated that there are more known forces on submarine hulls than surface vessels and that all the latest submarine-hull designs were first tested

ULTRASONICS IS THEIR SUBJECT

President M. B. Lunchick (left) of the Washington Area Section congratulates guest speaker S. D. Hart following his presentation of "A General Review of Ultrasonics" at the December 7th meeting. A National Bureau of Standards exhibit on methods of measurement is seen in the background



THEORY OF ELASTICITY

For engineers interested in calculating stresses in engineering structures and machine parts—here is the essential knowledge of the theory of elasticity, together with the solution of engineering problems of practical importance. The book discusses approximate and experimental methods of solving elastic problems which have proved useful in the study of complicated cases of stress distribution. **By S. Timoshenko and J. N. Goodier, Stanford Univ. 2nd Ed., 506 pp., 266 illus., $11.50**

THEORY OF PLATES AND SHELLS

Covers the general theory of bending of plates under lateral load, with applications to circular and rectangular plates, and with numerous tables simplifying the calculation of deflections and stresses in plates. The book provides a treatment of combined bending and tension or compression of plates, with applications to thin plates. Also dealt with is the theory of bending of shells, including applications to cylindrical and spherical shells. **By S. Timoshenko, Stanford Univ., and S. E. Woinowsky-Krieger, Université Laval, Quebec, Canada. 2nd Ed., 580 pp., 280 illus., $15.00**

FORMULAS FOR STRESS AND STRAIN

A compact and readily usable summary of important formulas, facts, and principles pertaining to strength of materials. This book is a reliable handbook for engineers concerned with machine and structural design which provides valuable formulas for stress analysis and elasticity. All such experimental data and empirical formulas are in line with current information. New material has been added, ensuring complete coverage of stress and strain. **By Raymond J. Roark, Univ. of Wisconsin. 3rd Ed., 381 pp., $8.25**

THE STRAIN GAGE PRIMER

What the bonded wire resistance strain gage can do in solving problems in experimental stress analysis is explained in this practical book. All phases are covered—from selecting the proper commercial gage through preparing surface of the test member, cementing, damage-proofing, and wiring the gage—to selecting the proper instrument to record the strain gage data, and interpreting readings in terms of significant stresses. **By C. C. Perry and H. R. Lissner, Wayne Univ. 280 pp., 180 illus., $6.00**

SEE THESE BOOKS 10 DAYS FREE

McGraw-Hill Book Co., Dept. EXM-2
327 W. 41st St., New York 36, N. Y.

Send me book(s) checked below for 10 days' examination on approval. In 10 days I will remit for book(s) I keep plus few cents for delivery costs, and return unwanted book(s) postpaid. (We pay delivery cost if you remit with this coupon—same return privilege.)

- ☐ Timoshenko & Goodier—**Theory of Elasticity, $11.50**
- ☐ Timoshenko & Woinowsky-Krieger—**Theory of Plates and Shells, $15.00**
- ☐ Roark—**Formulas For Stress & Strain, $8.25**
- ☐ Perry & Lissner—**Strain Gage Primer, $6.00**

Name .....
Address .....
City ..... Zone ..... State .....
Company .....
Position .....

For price and terms outside U. S. write McGraw-Hill Intl., N. Y. C. 36 **EXM-2**

and proved at the Model Basin.

The end of the talk was followed by a discussion period. The meeting was brought to a close by the program chairman, Everett Reid.

(Reported by Bernard P. Friesecke for the New England Section)

Pittsburgh

Science Fair Award

The Pittsburgh Section of the Society for Experimental Stress Analysis has established an annual award of twenty-five dollars ($25) to be presented as part of the Science Fair sponsored each year by the Buhl Planetarium in Pittsburgh.

This award will be presented to a selected entrant under the broad classification of "Experimental Mechanics." Each year the Awards Committee of the Pittsburgh Section will submit a detailed description of the exact field of endeavor upon which entries will be judged for consideration of the award. The committee will also assign judges to evaluate the entries and select the entrant to whom the award will be made.

(Reported by Meriden A. French for the Pittsburgh Section)

Washington Area

Ultrasonics and a Cantilever Bridge

The December 7th meeting was held in the Exhibit Room of the National Bureau of Standards, with Myron E. Lunchick presiding.

A number of business matters were discussed and voted upon, the most important of which was the unanimous adoption of a new Constitution and By-Laws which had been drafted by a committee under Roscoe L. Bloss. John Armstrong reported on the national meeting which had been held in Berkeley, and SESA Vice-president Irwin Vigness described some of the innovations introduced by the Society with regard to its publication policy, organizational structure, and dues.

The principal speaker of the evening was Stephen D. Hart of the U.S. Naval Research Laboratory, who lectured on "A General Review of Ultrasonics." Following a brief resumé of propagation theory, Mr. Hart described the types of wave motion, the principles of flaw detection, and the effects of such factors as attenuation, impedance and reflection. He went on to discuss the various types of transducers and related instrumentation, and methods of application. The presentation was concluded with a series of interesting demonstrations of flaw detection in metals and adhesive bonds.

A lively discussion of the subject followed, ranging from crack propagation in submarine hulls to automatic dish washing. The meeting was then recessed temporarily for coffee, doughnuts and good-fellowship (a Section tradition), and recovened shortly afterward for a showing of the U. S. Steel Co. film, "The Cantilever Bridge." The film depicted the erection of the nation's heaviest cantilever bridge at Tappan Zee on the Hudson River.

(Reported by L. Mordfin for Washington Area Section. Accompanying photo by N. H. Isey.)

Ideas on Starting a New Section

by Clarence R. Smith

In starting a new local section, like starting a new anything, the first question that arises is, what is the purpose? If there is no purpose, if you want to start a new section simply to be in on starting something new—forget it! Perhaps joining a lodge or little-theatre group would better satisfy your ambitions.

On the other hand, if there is a need for a local section of the Society for Experimental Stress Analysis in your area, perhaps you are the one to start the ball rolling. You know, you can't always leave it to others—it may never get done.

How do you know whether there is a need? You don't! One way to find out is to call a meeting and survey the response. Look through the membership list in Vol. XVII, No. 2, *Proceedings* and check off those living within a 150 mile radius. Do you know them? If you can find 20 members and personally are acquainted with less than 5, it is high time you were getting acquainted.

You might ask, "What do I get out of it?" Maybe nothing, If you give nothing, the chances are that you'll receive nothing! One of the purposes of the Society for Experimental Stress Analysis is the free interchange of ideas and experiences. How often have you worked hard on an assignment only to find that the answer was really simple? Perhaps someone has already solved a problem similar to the one you are now working just as frantically. You should get to know him.

Are you planning on buying a piece of equipment to do a certain job? Perhaps a new-found acquaintance at a local meeting could tell you how he solved a problem with similar apparatus in a much more economical way.

Can you or your company afford not to participate in a local section of the SESA? That depends. If you know all the answers and your company makes a product wherein there are no problems, then you would be wasting your company's time and money—and yours.

On the other hand, if you have problems, you may not have all the answers. For a very small investment in time, and perhaps the price of a dinner away from home, your association with others having similar interests will certainly stimulate you to more positive approaches in solving problems involving experimental mechanics.

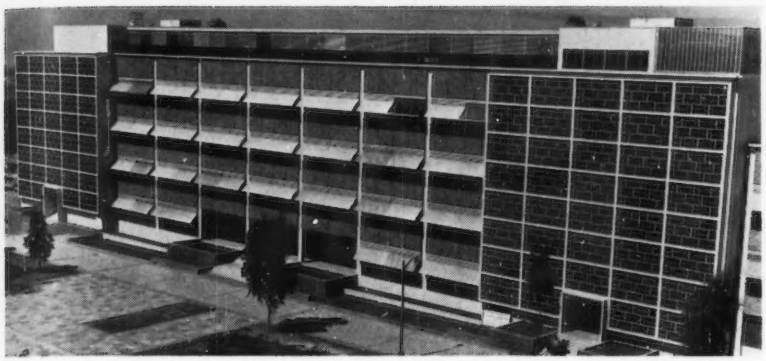
Assuming that you are ready to call a first meeting, you might first talk it over as to time and place. Colleges and universities are usually cooperative. Otherwise, there are many restaurants having private dining rooms available.

Your first meeting should feature a speaker or speakers capable of offering subject matter in keeping with local interests. Strain gages and photoelasticity usually create general interest. However, even these should introduce something timely. Perhaps someone in your area has already submitted a paper for presentation at a national meeting. There is absolutely no reason why he could not present it at a local meeting.

Do you want more information? If so write to me as chairman of the SESA Local Sections Committee. My home address is 1807 Beryl St., San Diego 9, Calif.

Clarence R. Smith, design specialist with Convair, Div. of General Dynamics Corp., San Diego, Calif. is chairman of the SESA Local Sections Committee.

from the campus



College of Engineering—University of California at Los Angeles

Plan Now To Attend
1961 Spring Meeting
May 10 - 12
Benjamin Franklin Hotel
Philadelphia, Pa.

FOURTH SYMPOSIUM ON ROCK MECHANICS

Both theory and practice will be covered by papers to be presented at the Fourth Symposium on Rock Mechanics, being held at Penn State on Mar. 30 to Apr. 1, 1961.

The program consists of four technical sessions on these subjects: (1) behavior of rock under static loading, (2) behavior of rock under dynamic loading, (3) measurement of rock properties and stress instrumentation and (4) ground control and subsidence. Papers to be presented include stress adjustment by roof control, behavior of rock beams under impact, new developments in physical properties measurement, photoelastic determination of stress adjacent to a cratered surface, stress waves from rock bursts, fracture patterns around underground openings, prestressed concrete pillars, stress distribution in rock beneath a drill bit, brittle fracture phenomena in rock and anchorage slippage in roof bolting.

The symposium is under the joint sponsorship of the departments of mining at Colorado School of Mines, University of Minnesota, Missouri School of Mines and Metallurgy and the Pennsylvania State University.

ENGINEERING TEACHER INCOME UP

The average total professional income of engineering teachers has increased by 14.7% in the past two years, according to a report by the Engineering Manpower Commission of Engineers Joint Council of New York City. The survey, made possible through a grant from The National Science Foundation, covers over 5000 engineering teachers or more than half the total number.

Seventy-seven and one-half percent of the average total income of engineering teachers is basic teaching salary and the remainder derived from outside professional work such as consulting and research. Since the survey was first made in 1956, average basic teaching salaries have risen 31.1% and income from "outside sources" 11.3%.

Today's average basic teaching salary for engineering teachers in American colleges is $8534. It is supplemented by an average of $2479 from outside professional income.

In public institutions the rise in total income (15.6%) was greater than that in private institutions (12.5%) due more to higher outside earnings than to a difference in the rise of basic teaching salaries. Such outside earnings for engineering teachers in public institutions increased 24.8% during the last two years as opposed to only 11.3% in private institutions. The basic teaching salary increase was 13.4% (public) compared with 13.0% (private).

Other statistics of the report show that salaries increase with age, academic rank and type of degree; that the median teaching salary for instructors is $5380, for full professors, $10,500.

"Salaries and Income of Engineering Teachers—1960" is the title of the report.

DATES OF FORTHCOMING SHORT COURSES

1. At the Southwest Research Institute, in San Antonio, Tex.
(Jointly sponsored with SESA)

Strain Gage Techniques	April 17–21, 1961

Detailed information will be available from Dr. M. M. Lemcoe, manager, Strength Analysis Section, Department of Structural Research, Southwest Research Institute, 8500 Culebra Rd., San Antonio, 6, Tex.

2. At the Massachusetts Institute of Technology, Cambridge, Mass.
(No sponsorship with SESA)

Industrial Photoelasticity	June 19–23
Non destructive Testing	June 26–30
Strain Gage Techniques (Lectures)	July 10–14
Strain Gage Techniques (Laboratory)	July 17–21

Detailed information will be available from Dr. W. M. Murray, Room 3-260, Massachusetts Institute of Technology, Cambridge, 39, Mass.

3. At the University of California, Los Angeles, Calif.
(Jointly sponsored with SESA)

Strain Gage Techniques (Lectures)	August 21–25
Strain Gage Techniques (Laboratory)	August 28–September 1
Industrial Photoelasticity	September 5–9

Detailed information will be available from Dr. Sam M. Houston, assistant head, Engineering Extension, Department of Engineering, University of California, Los Angeles, 24, Calif.

personnel

in the news

MAJOR MADE SALES MANAGER

Richard L. Major has joined the staff of Micro Systems, Inc., of San Gabriel, Calif., as sales manager. In this capacity, Mr. Major will be responsible for all sales and marketing activities in connection with the company's semiconductor strain gages, the radiation tracking transducer, and other solid-state products to be announced in the near future.

Prior to joining Micro Systems, Mr. Major was manager of technical services for Wiancko Engineering Co., responsible for sales through representatives. Before joining Wiancko, he was with Consolidated Electrodynamics, and prior to that, with Schlumberger Well Surveying Corp.

Major is a 1950 graduate of the University of Southern California.

APPOINTED DISTRICT MANAGER BY HUGHES

Thomas S. Burns has been appointed manager of the Detroit, Mich., district sales office of Hughes Aircraft Co.'s semiconductor division.

Before joining Hughes, Mr. Burns was a district sales manager for the General Electric Co. He was graduated from the United States Naval Academy in 1951 with a bachelor of science degree in electrical engineering. He received a master's degree in business administration from the University of Massachusetts in 1958.

Mr. Burns is a lieutenant in the U. S. Naval Reserve.

WRIGHT PROMOTED

William V. Wright has been named a vice-president of Electro-Optical Systems, Inc. of Pasadena, Calif. Dr. Wright will continue in his present position as manager of the company's Solid State Division; however, he will also acquire responsibilities for policies concerning proprietary programs, subsidiaries, and expansion activities.

Dr. Wright has been with Electro-Optical Systems since 1957, and as manager of the Solid State Division has been responsible for directing the company's activities in solid state device and materials research.

Prior to joining EOS, Dr. Wright was program director, semiconductor materials, for Pacific Semiconductors, Inc., where he was active in all phases of semiconductor technology.

WOLFE JOINS BORG-WARNER CONTROLS

C. M. Wolfe has joined Borg-Warner Controls as head of transducer engineering.

A specialist in instruments and devices, Dr. Wolfe will supervise transducer research and development programs. These programs will include refinements in existing types of transducers, as well as new product developments and applications.

Dr. Wolfe has spent 16 years in research and development in the fields of missile systems, rocket motors and instrumentation. Additional experience includes two years in aerodynamic analysis and seven years in teaching at the University of Washington. He holds B.S., M.S. and Ph.D. degrees in electrical engineering from California Institute of Technology.

DARVEAU NAMED SALES ENGINEER

George F. Darveau has been named sales engineer for Statham Instruments, Inc., of Los Angeles, Calif.

Mr. Darveau will serve clients in Illinois, Indiana, Wisconsin, Iowa, Minnesota, Nebraska, North Dakota and South Dakota, from offices in Metropolitan Chicago. The office address is 643 N. Emerald Ave., Mundelein, Ill.

Mr. Darveau, who has been in instrument sales for the past six years, comes to Statham from International Register Co.

A native of Gardiner, Me., Darveau received his B.A. degree from the University of Maine.

TRAGESSER DISTRIBUTION MANAGER

Herbert R. Tragesser has been named distribution manager at Non-Linear Systems, Inc.

Mr. Tragesser will be responsible for distribution of NLS digital voltmeters and digital measuring instruments used throughout the world. He will also head NLS's customer service department at the Del Mar factory.

Mr. Tragesser has had 13 years' experience in the electrical engineering field and is a graduate in electrical engineering of the University of Cincinnati.

ADMIRAL SCHERINI JOINS AIRPAX

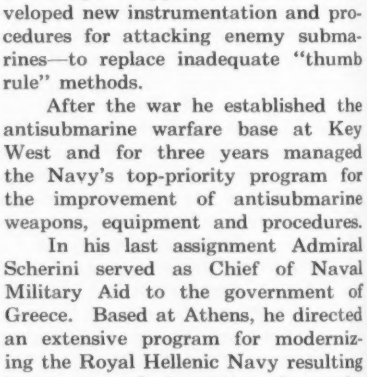
The board of directors of Airpax Electronics has elected Rear Admiral Otto A. Scherini a vice-president of the corporation, in charge of marketing.

Admiral Scherini is a graduate of the Naval Academy, class of 1932. His naval career has included numerous top positions in a variety of assignments, with emphasis on antisubmarine warfare.

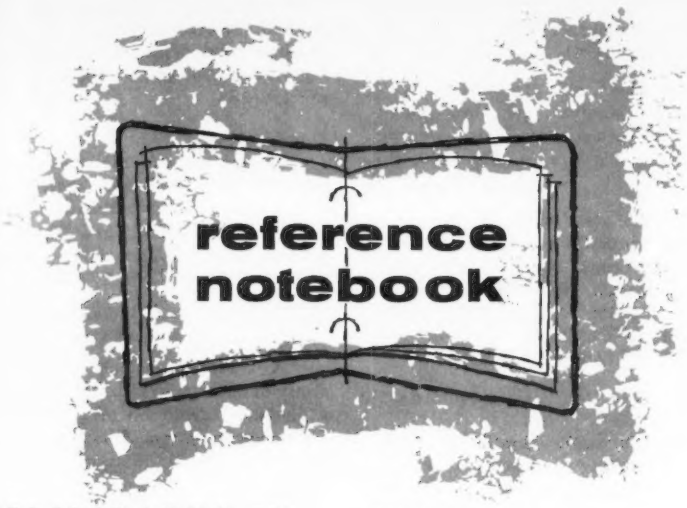
It was during World War II, as a destroyer skipper, that he first developed new instrumentation and procedures for attacking enemy submarines—to replace inadequate "thumb rule" methods.

After the war he established the antisubmarine warfare base at Key West and for three years managed the Navy's top-priority program for the improvement of antisubmarine weapons, equipment and procedures.

In his last assignment Admiral Scherini served as Chief of Naval Military Aid to the government of Greece. Based at Athens, he directed an extensive program for modernizing the Royal Hellenic Navy resulting in a personal decoration from the King of Greece.



Otto A. Scherini



BIBLIOGRAPHY ON THREE-DIMENSIONAL PHOTOELASTICITY 1949–1959

Compiled by
Maurice H. Smith and Mary Lee

Continued from page 34A of January 1961 issue of Experimental Mechanics

PUBLISHED PAPERS AND RESEARCH REPORTS

Barat, A.

La photoélasticimétrie. *Machines et Métaux*, v. 31, No. 350, October 1947, pp. 333–339, 341.

Barnhard, K. E., Jr., Hale, A. L., and Meriam, J. L.

Stresses in rotating disks due to noncentral holes. *Society for Experimental Stress Analysis. Proceedings*, v. 9, No. 1, 1951, pp. 35–52.

Barwell, F. T.

"Frozen stress" in molded polystyrene. *International Rheological Congress*, Holland, 1948. *Proceeaings*, v. 2, pp. 232–238; v. 3, pp. 65–66.

Barriage, J. B., and Durelli, A. J.

Application of a new deformeter to two-dimensional thermal stress problems. *Society for Experimental Stress Analysis. Proceedings*, v. 13, No. 2, 1956, pp. 97–106.

Bell, J. F.

Normal incidence in the determination of large strain through the use of diffraction gratings. *U. S. National Congress of Applied Mechanics, 3rd Proceedings*, June 1958, pp. 489–493.

Bell, J. F.

Determination of dynamic plastic strain through the use of diffraction gratings. *Journal of Applied Physics*, v. 27, October 1956, pp. 11-9-1113.

Benito, C., and Moreno, A.

Three-dimensional photoelastic studies with gelatin models (in Spanish). *Laboratorio Central de Ensayo de Materiales Construccion.* Madrid. *Publication* No. 73, 1951, 34 pp.

Betser, A. A., Flynn, P. D., and Frocht, M. M.

On the stress-optic law under impact loading. *International Congress of Applied Mechanics*, Brussel 1957. *Proceedings*, v. 8, pp. 367–377.

Betser, A. A., and Frocht, M. M.

A photoelastic study of maximum tensile stresses in simply supported short beams under central transverse impact. *Journal of Applied Mechanics*, v. 24, Dec. 1957, pp. 5-9-514.

Bonvalet, C., and Gobulovie, G.

Étude comparatice de l'état de contraintes des eprauvettes de résilience, etc. *Annales de l'Institute Technique du Batiment et des Travaux Publics*, v. 9, No. 106, 1956, pp. 899–921.

Brock, J. S.

The determination of effective stress and maximum shear stress by means of small cubes taken from photoelastic models. *Society for Experimental Stress Analysis. Proceedings*, v. 16, No. 1, 1958, pp. 1–8.

Brown, A. F. C.

Triaxial stresses caused by notches. *British Journal of Applied Physics*, v. 5, No. 8, August 1954, pp. 280–284.

Brown, A. F. C., and Hickson, V. M.

A photoelastic study of stresses in screw threads. *Institution of Mechanical Engineers. Proceedings*, (B) 1B, 12, 1952/53, pp. 605–608.

Brown, A. F. C., and Hickson, V. M.

Photoelastic Laboratory at National Physical Laboratory. *Engineering*, v. 171, No. 4454, June 1951, pp. 701–704.

Brown, A. F. C.

Measurement of the curvature of stress trajectories in photoelastic models. *British Journal of Applied Physics*, v. 2, No. 5, May 1951, pp. 138–140.

Brown, A. F. C., and Hickson, V. M.

Improvements in photoelastic technique obtained by the use of a photometric method. *British Journal of Applied Physics*, v. 1, Feb. 1950, pp. 39–44.

Christodoulides, S. L.

The distribution of stresses around the end anchorages of prestressed concrete beams. Comparison of results photoelastically with strain gage measurements and theoretical solutions. *International Association of Bridge and Structures Engineers. Publications*, v. 16, 1956, pp. 55–70.

Christodoulides, S. L.

Use of photoelasticity in stress analysis. *Civil Engineering* (London), v. 50, No. 591, September 1955, pp. 984–986.

Clutterbuck, Margery

The dependence of stress distribution on elastic constants. *British Journal of Applied Physics*, v. 9, August 1958, pp. 323–329.

Cole, A. G., and Brown, A. F.

Photoelastic determination of stress concentration factors caused by a single U-notch on one side of a plate in tension. *Royal Aeronautical Society, Journal*, v. 62, August 1958, pp. 597.

Craggs, J. W.

Characteristic surfaces in ideal plasticity in three-dimensions. *Quarterly Journal of Mechanics and Applied Mathematics*, v. 7, part 1, March 1954, pp. 35–39.

D'Agostino, J., Drucker, D. C., and others

Epoxy adhesives and casting resins as photoelastic plastics. *Brown University, Graduate Division of Applied Mathematics, Report* No. DA-798/10, April 1954, 9 pp. (ASTIA AD-29,058).

Dally, J. W., Riley, W. F., and Durelli, A. J.

A photoelastic approach to transient stress problems employing low modulus materials. *Air Force Office of Scientific Research.* TN-58-891, December 1958, 32 pp. (ASTIA AD-

Based on Literature Search No. 17 published in April 1960 by the The James Forrestal Research Center Library, Princeton University, Princeton, N. J.

204,138). (Armour Research Foundation. Contract AF 18(603)144.)

Dally, J. W., Durelli, A. J., and Riley, W. F.
Photoelastic study of stress wave propagation in large plates. *Air Force Office of Scientific Research.* AFOSR TN 58-748, November 1958, 28 pp. (ASTIA AD 162,271). (Armour Research Foundation, Illinois Institute of Technology.)

Dally, J. W., Durelli, A. J., and Riley, W. F.
A new method to "lock-in" elastic effects for experimental stress analysis. *Journal of Applied Mechanics*, v. 25, June 1958, pp. 189–195.

Drucker, D. C., and Woodward, W. B.
Interpretation of photoelastic transmission patterns for three-dimensional model. *Journal of Applied Physics*, v. 25, No. 4, April 1954, pp. 510–512.

Drucker, D. C.
The method of oblique incidence in photoelasticity. *Society for Experimental Stress Analysis. Proceedings*, v. 8, No. 1, 1950, pp. 51–66.

Drucker, D. C., and Frocht, M. M.
Equivalence of photoelastic scattering patterns and membrane contours for torsion. *Society of Experimental Stress Analysis. Proceedings*, v. 5, No. 2, 1948, pp. 34–41.

Drucker, D. C., and Mindlin, R. D.
Stress analysis by three-dimensional photoelastic methods. *Journal of Applied Physics*, v. 11, 1940, pp. 724–732.

Durelli, A. J., and Riley, W. F.
Experimental means of determining wave propagation phenomena in solids. *Armour Research Foundation.* February 1959. 15 pp., 17 references. (AFOSR TR-59-21.) (Contract AF 18(603)144) (ASTIA AD-211,667).

Durelli, A. J., Dally, J. W., and Riley, W. F.
Stress concentration factors under dynamic loading conditions. *Air Force Office of Scientific Research.* AFOSR TN 58-892, December 1958, 23 pp. (ASTIA AD-204,137.) (Armour Research Foundation, Illinois Institute of Technology). (Contract AF 18(603)-144.)

Durelli, A. J., Dally, J. W., and Riley, W. F.
Stress and strength studies on turbine blade attachments. *Society for Experimental Stress Analysis. Proceedings*, v. 16, 1958, pp. 171–182.

Durelli, A. J., and Kobayashi, A. S.
Stress distributions around hydrostatically loaded circular holes in the neighborhood of corners. *Journal of Applied Mechanics*, v. 25, June 1958, pp. 178–183.

Durelli, A. J.
Experiments for determination of transient stress and strain distributions in two-dimensional problems. *American Society of Mechanical Engineers—Transactions.* (*Journal of Applied Mechanics*) v. 24, No. 1, March 1957, pp. 69–76.

Durelli, A. J.
A three-dimensional photoelastic study of contact stresses in the head of a model of a railroad rail. *Society for Experimental Stress Analysis. Proceedings*, v. 14, No. 1, 1956, pp. 11–12.

Durelli, A. J., and Barriage, J.
Stress distribution in square plates with hydrostatically loaded central circular holes. *Journal of Applied Mechanics*, v. 22, No. 4, December 1955, pp. 539–544.

Durelli, A. J., and Riley, W. R.
Stress distribution in strips with hydrostatically loaded central circular holes. *Midwestern Conference on Solid Mechanics, 2nd Proceedings,* October 1955, pp. 81–93.

Durelli, A. J., and Tsao, C. H.
Determination of thermal stresses in three-ply laminates. *Journal of Applied Mechanics*, v. 22, 1955, pp. 190–192; discussion, p. 611.

Durelli, A. J., and Barriage, J.
Stress distribution in square plates with hydrostatically loaded central circular holes. *Journal of Applied Mechanics*, v. 22, 1955., pp. 539–544; discussion, v. 23, 1956, pp. 325–326.

Durelli, A. J., and Phillips, E. A.
Criteria for selecting experimental stress analysis methods. *Product Engineering*, v. 26, No. 1, January 1955, pp. 182–191.

Durelli, A. J., Lake, R. L., and Okubo, S.
A grid method to determine strains in rubber-like material. *U. S. National Congress of Applied Mechanics, 2nd,* June 1954; *Proceedings*, pp. 219–295.

Durelli, A. J., Lake, R. L., and Tsao, C. H.
Device for applying uniform loading to boundaries of complicated shape. *Society for Experimental Stress Analysis. Proceedings*, v. 11, No. 1, 1953, pp. 55–61.

Durelli, A. J., Lake, R. L., and Phillips, E.
Stress concentrations produced multiple semi-circular notches in infi-

BOUND VOLUMES OF

PROCEEDINGS AVAILABLE

Still available are a number of bound volumes of the series **Proceedings of the Society for Experimental Stress Analysis.** Published twice yearly, these volumes comprise the technical papers presented at national meetings and contain the results and descriptions of analytical and experimental investigations, as well as discussions of technique and instrumentation studies.

Price per copy: $7.00 to nonmembers and $6.00 to members; plus postage. Copies may be ordered through the Society for Experimental Stress Analysis, 21 Bridge Square, Westport, Connecticut.

nite plates under uniaxial state of stress. *Society for Experimental Stress Analysis. Proceedings*, v. 10, No. 1, 1952, pp. 53–64.

Durelli, A. J., and Lake, R. L.

Some unorthodox procedures in photoelasticity. *Society for Experimental Stress Analysis. Proceedings*, v. 1, No. 1, 1951, pp. 97–122.

Durelli, A. J., Lake, R. L., and Phillips, E.

Stress distribution in plates under a uniaxial state of stress, with multiple semicircular and flat-bottom notches. *U. S. National Congress of Applied Mechanics*, 1951, *Proceedings*, pp. 309–315.

Durelli, A. J., and Lake, R. L.

Three-dimensional photoelasticity. *Machine Design*, v. 22, Dec. 1950, pp. 122–125.

Durelli, A. J.

Experimental determination of the isostatic lines. *ASME Transactions*, v. 9, 1942, pp. A-155-A-160 (*Journal of Applied Mechanics*).

Durelli, A. J., and Murray, W. M.

Stress distribution around a circular discontinuity in any two-dimensional systems of combined stress. *Eastern Photoelasticity Conference*, 14th Semi-Annual. Yale University, December 1941, *Proceedings*.

Favre, H., and Gilg, B.

Sur une méthode purement optique pour la mesure directe des moments dans les plaques minces fléchies. *Schweizerische Bauzeitung*, v. 68, Nos. 19–20, May 1950, pp. 253–257, 265–267.

Fessler, H.

"Frozen stress" phenomenon in photoelasticity. *Institution of Mechanical Engineers. Proceedings*, v. 1-B, No. 12, 1952/1953, pp. 613–620.

Fisher, A. P., and Heywood, R. B.

Freezing of stresses. *Engineer*, v. 184, No. 4795. December 1947, pp. 586–587. (See also *Engineering*, v. 165, No. 4276, January 1948.)

Fisher, V. A. P.

"Basic physical properties relied upon in the frozen stress technique," *Institution of Mechanical Engineers. Proceedings*, v. 158, No. 2, 1948, pp. 230–235, 2 plates.

Flügge, W.

Statics of the wing. (In French.) *France. Office National d'Études et Recherches. Aéronautique.* Rapport No. 24, 1949, 52 pp.

Flanagan, James H.

The single load calibration of photoelastic plastics. *Watertown Arsenal. Ordnance Materials Research Office. Report* No. 4 (Project TB 4-911, April 1955, 7 pp. (ASTIA AD-68,993.)

Flanagan, J. H.

A discussion of the photoelastic isochromatic negative. *Watertown Arsenal Ordnance Materials Research Office. Report* No. WAL 893/163, Jan. 15, 1954. 18 pp., 10 references. (ASTIA AD 32,964.)

Föppl, L.

Examples of application of photoelasticity. (In German.) *VDI Zeitschrift*, v. 93, No. 5, February 1951, pp. 105–112.

Forsyth, P. J. E..

A study of fatigue crack formation in silver chloride. *Gt. Brit. Royal Aircraft Establishment. Technical Note* No. Met. 276, November 1957, 19 pp. (ASTIA AD-200,823.)

Fox, E. A.

The photoelastic determination of stress on transverse planes of symmetry for the general axi-symmetric case. *Columbia University.* (Contract Nonr-26609.) *Technical Report* No. 15, June 1954, 8 pp. 11 references. (ASTIA AD-35,967.)

Franz, W. F.

A three-dimensional photoelastic stress analysis of a threaded drill pipe joint. *Society for Experimental Stress Analysis. Proceedings*, v. 9, No. 2, 1952, pp. 185–194.

Fried, B., and Shoup, N. A.

A study in photoplasticity; the photoelastic effect in the region of large deformations in polyethylene. *Washington State College. Division of Industrial Research. Technical Report* No. 3, May 1953, 6 pp. (ASTIA AD-12,803). (Contract N7onr-330, T. O.3.)

Fried, B.

Some observations on photoelastic materials stressed beyond the elastic limit. *Society for Experimental Stress Analysis. Proceedings*, v. 8, No. 2, 1951, pp. 143–148.

Frocht, M. M. and Flynn, P. D.

On Saint Venant's principle under dynamic conditions. *Illinois Institute of Technology, Laboratory for Experimental Stress Analysis. Technical Report* No. 8, January 1959, 10 pp. (Contract DA 11-022-ORD-1609.) (ASTIA AD 211,351.)

Frocht, M. M., and Thomson, R. A.

A three-dimensional photoelastic investigation of the stress distribution in an axially thick ring under diametral loads. *Illinois Institute of Technology. Laboratory of Experimental Stress Analysis. Technical Report* No. 7, April 1958, 26 pp. (ASTIA AD 202,243.)

PROFESSIONAL OPPORTUNITIES IN STRUCTURES ENGINEERING AT AEROJET GENERAL'S SOLID ROCKET PLANT

In the sixties, as in the forties, Aerojet-General stands foremost in the solid rocket field. Our Solid Rocket Plant near Sacramento, California, is the nation's largest facility for the development and production of solid rocket power plants.

To continue our growth and expansion into new fields of rocketry, we are seeking mature, experienced engineers who have specialized in the areas of structural analysis, shell theory, large-deformation theory, heterogeneous construction (such as filament-winding), photoelastic analysis, and quantitative structural testing. A degree in mechanical, aeronautical or civil engineering is required.

The intricate and highly refined load-carrying components of moderr missiles present interesting challenges to our structural engineering staff. Intriguing problems are brought about by the complex interaction of the very large propellant grain, chamber, interstage structures and applied loads which a modern large solid rocket motor must withstand. These problems demand the attention of structural analysts and test engineers of the highest talent.

Our top specialists are given responsibility and freedom to work on a broad spectrum of problems. Excellent salaries are offered, commensurate with proven ability and experience. Our location near Sacramento, between Lake Tahoe and San Francisco, offers the finest living conditions.

We invite you to send your resume and any detailed questions to:

Mr. E. P. James, Head
Technical and Scientific Personnel
Box 1947-JJ

AEROJET-GENERAL CORPORATION
Sacramento, California

Frocht, M. M. and Thomson, R. A.
Studies in photoplasticity. *U. S. National Congress of Applied Mechanics, 3rd Proceedings*, 1958, pp. 553–540.

Frocht, M. M., and Thomson, R. A.
Studies in photoplasticity. *Illinois Institute of Technology. Laboratory of Experimental Stress Analysis. Technical Report* No. 5, December 1957, 15 pp. (ASTIA AD-149,863.)

Frocht, M. M. and Srinath, L. S.
Non-destructive method for three-dimensional photoelasticity. *U. S. National Congress of Applied Mechanics, 3rd*, 1958. *Proceedings*, pp. 329–337.

Frocht, M. M. and Srinath, L. S.
A non-destructive method for three-dimensional photoelasticity. *Illinois Institute of Technology. Laboratory of Experimental Stress Analysis. Technical Report* No. 6, Dec. 1957, 12 pp. (ASTIA AD 149,862.)

Frocht, M. M., Flynn, P. D., and Landsberg, D.
Dynamic photoelasticity by means of streak photography. *Society for Experimental Stress Analysis. Proceedings*, v. 14, No. 2, 1957, pp. 81–90.

Frocht, M. M., and Betser, A. A.
Photoelastic study of maximum tensile stresses in simply supported short beams under central transverse impact. *ASME-TRANS.* (*J. Applied Mechanics*), v. 24, No. 4, December 1957.

Frocht, M. M.
Photoelastic study of contact stresses in head of model of railroad rail. *Society for Experimental Stress Analysis*, v. 14, No. 1, *Proceedings*, 1956.

Frocht, M. M.
A three-dimensional photoelastic study of contact stresses in the head of a model of a railroad rail. *Society for Experimental Stress Analysis. Proceedings*, v. 14, No. 1, 1956, pp. 1–10.

Frocht, M. M. and Flynn, P. D.
Studies in dynamic photoelasticity. *Journal of Applied Mechanics*, v. 23, 1956, pp. 116–122; discussion, pp. 482–484, by A. J. Durelli.

Frocht, M. M., and Guernsey, Roscoe Jr.
Further work on the general three-dimensional photoelastic problem. *Journal of Applied Mechanics*, v. 22, 1955, pp. 183–189; discussion, v. 23, 1956, pp. 134–136, by A. J. Durelli and J. M. Barriage.

Frocht, M. M., Pih, H., and Landsberg, D.
The use of photometric devices in the solution of the general three-dimensional photoelastic problem. *Society for Experimental Stress Analysis. Proceedings*, v. 12, No. 1, 1954, pp. 181–191.

Frocht, M. M., and Guernsey, R., Jr.
A special investigation to develop a general method for three-dimensional photoelastic stress analysis. *NASA Report* 1148, Nov. 1953, 17 pp. (Supersedes *NACA TN* 2822, 59 pp., December 1952.)

Frocht, M. M., Guernsey, R., Jr., and Landsberg, D.
Photoelasticity—a precision instrument of stress analysis. *Society for Experimental Stress Analysis. Proceedings*, v. 11, No. 1, 1953, pp. 105–114.

Frocht, M. M., Guernsey, R., Jr., and Landsberg, D.
A photoelastic re-examination of notched tension bars, *Journal of Applied Mechanics*, v. 19, No. 1, March 1952, pp. 124.

Frocht, M. M., and Landsberg, D.
Factors of stress concentration in bars with deep sharp grooves and fillets in tension. *Society for Experimental Stress Analysis. Proceedings*, v. 8, No. 2, 1951, pp. 149–162; discussion by A. J. Durelli and R. H. Jacobson, pp. 163–170.

Frocht, M. M., and Guernsey, R., Jr.
Studies in three-dimensional photoelasticity. *U. S. National Congress of Applied Mechanics. Proceedings*, June 1951, pp. 301–307.

Frocht, M. M.
On the removal of time stresses in three-dimensional photoelasticity. *Society of Experimental Stress Analysis. Proceedings*, v. 5, No. 2, 1948, pp. 9–13.

Frocht, M. M.
Studies in three-dimensional photoelasticity torsional stresses by oblique incidence. *ASME Transactions*, v. 11, 1944, pp. A-229-234. (*Journal of Applied Mechanics.*)

Frocht, M. M.
Studies in three-dimensional photoelasticity. Stress concentrations in shafts with transverse circular holes in tension. *Journal of Applied Physics*, v. 15, 1944, pp. 72–88.

Frocht, M. M.
Studies in three-dimensional photoelasticity. Stresses in bent circular shafts with a transverse holes—correlation with results from fatigue and strain measurements. *ASME Transactions*, v. 11, 1944, pp. A-10-16; discussion, p. A-253-254, by D. C. Drucker. (*Journal of Applied Mechanics*) also *Society for Experimental Stress Analysis. Proceedings*, v. 2, No. 1, 1944, pp. 128–138.

(*Continued Next Month*)

SPEAKERS SOUGHT BY SESA SECTIONS

Several requests have been received at SESA headquarters for names of qualified speakers to address local section meetings on subjects of current interest to members. Because of small budgets, a speaker's fee is not contemplated other than travel expense, where necessary.

Speakers willing to address SESA groups under these conditions are urged to so advise the Society for Experimental Stress Analysis, 21 Bridge Square, Westport, Conn. Please give preferred topic and length of talk. Also, if possible, send a brief abstract.

SOCIETY FOR EXPERIMENTAL STRESS ANALYSIS

21 Bridge Square, Westport, Connecticut

Individual Membership

Individual membership is intended for those who wish to keep abreast of the developments in the field of experimental stress analysis. Listed among the members are engineers, metallurgists, scientists, professors, consultants, students and laboratory personnel.

ANNUAL DUES	$10 up to June 30, 1961 $15 beginning July 1, 1961

Principal Benefits:

1. Annual subscription to EXPERIMENTAL MECHANICS, monthly journal of the SESA.
2. Special membership rates for the purchase of all technical publications of the SESA.
3. Reduced registration fee when attending Society meetings.

Membership Lapel Pins

A membership lapel pin bearing the SESA insignia attractively set in gold and blue is available for purchase by members. The price is $5.00, including postage.

Corporate Membership

Corporate membership is intended for those companies within whose laboratories there exists a group interest in stress analysis, for those who wish to encompass separate departments and several individuals under one membership or for the company wishing to avail itself of the many benefits open to corporate members. In some instances the information dealing with techniques developed in other laboratories, and made available by the Society, has enabled corporate members to avoid unnecessary duplication of research, thereby effecting savings many times greater than the cost of membership dues. Companies carrying this type of membership help support the many services the Society renders to industry and the individual.

ANNUAL DUES	$100.00

Principal Benefits:

1. Annual subscription to two copies of EXPERIMENTAL MECHANICS.
2. One set of bound PROCEEDINGS, with company's name imprinted in gold on front cover.
3. Reduced rates for the purchase of all technical publications of the SESA.
4. Substantial reduction on the regular rate for exhibiting equipment at National Meetings.

Application for Membership in

SESA

Society for Experimental Stress Analysis

Date_____

I hereby apply for national membership in the Society for Experimental Stress Analysis for the period starting on _____ and ending on June 30, 19____.

Dues in the amount of $_____, covering this period, are attached hereto. Also, the required information concerning my personal background and professional qualifications is given below for your consideration.

NAME_____
LAST FIRST MIDDLE

HOME ADDRESS_____
STREET

_____
CITY ZONE STATE COUNTRY, IF NOT USA

BUSINESS ADDRESS_____
COMPANY DEPARTMENT

_____
STREET

_____
CITY ZONE STATE COUNTRY, IF NOT USA

Application continues on other side ▶

(Please type or print all information. Mail completed application to Executive Secretary, Society for Experimental Stress Analysis, 21 Bridge Square, Westport, Conn.)

Meetings

The Society for Experimental Stress Analysis holds two major meetings each year. These meetings are scheduled each Spring and Fall, and take place at various locations throughout the country. The programs usually include:

1. **Technical Sessions**—Oral presentations and discussions outlining new equipment, new techniques and suggested improvements for testing and analyzing the strength of materials, structures and machine parts. Both members and nonmembers are invited to submit papers to the Society for consideration toward presentation and publication. "An SESA Paper," which outlines the procedure for the submission of manuscripts, as well as a copy of the abbreviated "Manuscript Check List," may be obtained from the Society upon request.
2. **Exhibits**—Commercial exhibits display the most up-to-date electronic equipment, testing apparatus and instruments in use by industry and research laboratories. Educational exhibits are designed to demonstrate and explain the scientific principles and techniques of experimental methods in stress analysis.
3. **Educational Lectures**—These serve to introduce the newcomer to the basic applications of photoelasticity, stress-coat and strain gages, together with their allied instrumentation and techniques.
4. **Plant Tours**—Group tours to plants in the area of the meeting offer first-hand inspection of industrial application of experimental stress analysis.

Local Sections

Throughout the U. S. there are local SESA sections which plan their activities in much the same manner as the parent organization. They schedule meetings, technical programs, educational programs, lectures, plant tours and, in some sections, publish their own newsletter. Local sections assist in the preparation of Spring and Fall Meetings and arrange for the many functions that take place during such meetings. Participation in local section activities allows frequent association among individuals who have a common interest in this particular field and offers an excellent opportunity for the interchange of information on the latest developments in stress analysis. A local representative will be glad to give you further details. Please direct all inquiries to the SESA Headquarters.

Special Services

In addition to the activities mentioned above, the Society has expanded its facilities to included other services, as follows:

1. **Educational Opportunities**—A notable example of the Society's educational activity is represented by the Special Summer Program on Strain Gage Techniques, sponsored jointly by the Society and major universities.
2. **Reprints** of papers published in EXPERIMENTAL MECHANICS may be purchased within 3 months after publication of issue in question. Further details available upon request.
3. **Opportunities to Students**—In order to encourage the participation of engineering students in SESA activities, the Society extends discount privileges to students who wish to attend SESA meetings. Special consideration is also extended in the purchase of SESA publications for class study.

SESA Membership Application Form

start on other side

TITLE OR POSITION_____

ADDRESS TO BE USED FOR MAIL: HOME ☐ COMPANY ☐

PRINCIPAL FIELD OF INTEREST_____

DATE OF BIRTH_____ PLACE OF BIRTH_____ CITIZEN OF_____ (IF NOT USA)

TECHNICAL EDUCATION:

	COLLEGE OR UNIVERSITY	LOCATION	DEGREE	DATE	COURSE
1.					
2.					
3.					

FORMER NATIONAL MEMBERSHIP IN SESA: YES ☐ NO ☐ _____ IF YES, LATEST YEAR

PRESENT AFFILIATION WITH SESA LOCAL SECTION: YES ☐ NO ☐ _____ NAME

SESA SECTION AFFILIATION PREFERRED_____

RECOMMENDED FOR MEMBERSHIP BY_____

OTHER SOCIETY AFFILIATIONS_____ NAME AND GRADE OF MEMBERSHIP

SIGNATURE_____

(Please type or print all information. Mail completed application to Executive Secretary, Society for Experimental Stress Analysis, 21 Bridge Square, Westport, Conn.)

STRAIN GAGE INSTRUMENTS

NEW

BA-3 BAM-1 without large meter. In a case which racks 6 in an 8¾" x 19" panel. Or makes a neat package for individual use. Has own batteries. DC system—dc to 20,000 cps flat. Drives tape recorders and scopes directly. —$600

A-2 Output amplifier. Same case as BA-3. Own batteries. DC to 10,000 cps flat. Gain 2-20. Drives all galvos. Max. output ±60 ma into 50 ohms. Designed to be driven by BA-3, BAM-1, BAM-3. —$400

BA-3

A-2

REGULARS

BAM-1 General purpose box. Static on own meter. Dynamic with 'scope. DC bridge, dc transistor amplifier, 2 or 4 arm bridge. Battery operated. Low amplifier noise and drift. Frequency dc to 20,000 cps. Differential input cancels hum. —$525

B-1 Bridge. No amplifier. Operates off 115v 60 cycle line. 100 volts at 75 ma. max dc bridge power. 1% regulation. Line noise satisfactory on static indicators, differential input oscilloscopes and floating galvos. Not satisfactory on single sided scopes. Frequency dc to 20,000 cps. —$425

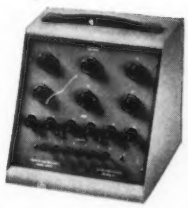
BA-12 Monitors fatigue tests. "Dynamic null balance" system gives wheatstone bridge accuracy to measurement of steady state dynamic signals including dc offset. Uses 'scope as indicator. 5 to 20,000 cps. —$650

BSG-6 6 channel switch and balance. Bridge power attenuation on each channel. 1, 2 or 4 external bridge arms. Any compensating gage can be connected into one or more channels. —$350

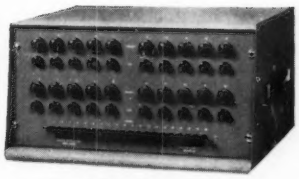
BSG-20 20 channel switch and balance. 8¾" x 19" rack panel size. —$900

BAM-1

B-1



BSG-6



BSG-20

COMING

BA-13 New version of BA-12. Transistor amplifier. Frequency 1 to 20,000 cps. Variable SPREAD pot as well as fixed values. MEASURE zero at end of dial with + − switch. BAM-1 case. —around $800

BAM-3 Transducer box. Similar to BAM-1 but with minimum confusion of controls. 4 arm bridges only. Zero left meter. Definite ranges selected by switch. 'Scope connection. Frequency dc to 10,000 cps. —around $500

ELLIS ASSOCIATES

67 Lincoln Avenue, Pelham, New York • Telephone 914-PE 8-3360

SALES REPRESENTATIVES

Michigan
S. Sterling Company
15310 W. McNichols Rd.
Detroit 35, Michigan
Broadway 3-2900

Ohio
S. Sterling Company
5827 Mayfield Road
Cleveland 24, Ohio
Hillcrest 2-8080

Indiana
Pivan Engineering Co.
10302 N. College Ave.
Indianapolis 20, Ind.
Victor 6-1532

Illinois, Wisconsin
Pivan Engineering Company
3542 W. Peterson Avenue
Chicago 45, Illinois
Keystone 9-4838

Canada
Tatnall Measuring & Nuclear Systems Ltd.
17 Donway West
Don Mills, Ontario
Plymouth 5-5122

All other areas—Write or phone collect to home office.

For details, circle No. 10 on Reader Information Card

PORTABLE

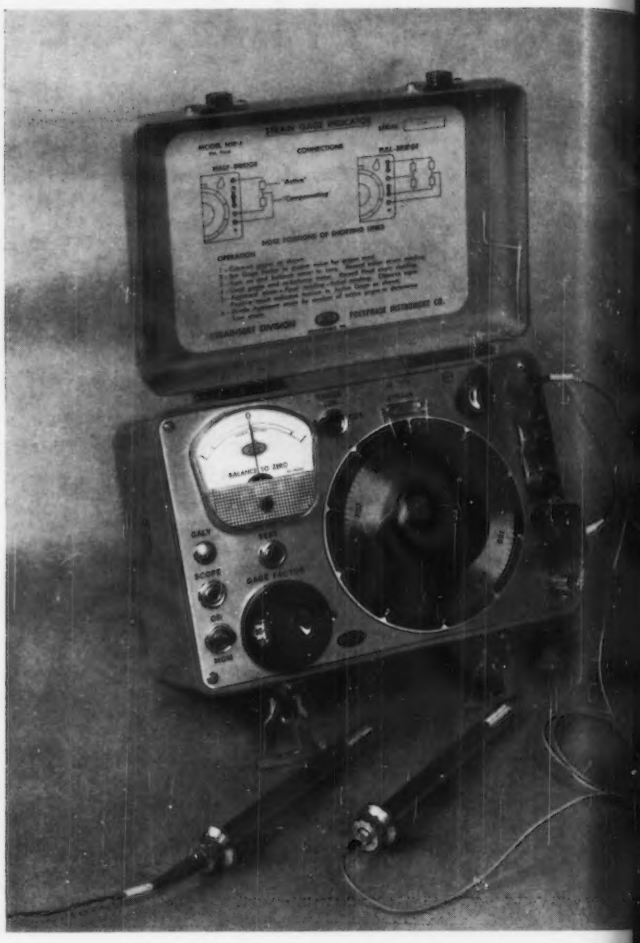
STRAIN INDICATOR

COMPLETELY TRANSISTORIZED

MODEL HW-1*

You Can Now—

- *Use 500-ft Long Lead Wires*
  With No Sacrifice In Sensitivity or Need for Phase Balancing!
- *Carry It Anywhere*
  It Weighs 6½-Lbs Only and is 6-in. x 6-in. x 9-in. Overall!
- *Save Real Money*
  It Costs $495.—Only!
  Battery Life 100-Hours!



SPECIFICATIONS:

Circuits: Unique circuitry renders instrument insensitive to capacitance or phase unbalance due to long lead wires.
Range: ±30,000 micro inches/inch; 60,000 micro inches/inch total.
Gage Factor Range: From 1.50 to 4.50.
Gage Resistance Range: 50 ohms to 2,000 ohms.
Readability: ±1 Micro inch/inch.
Sensitivity: Meter is sensitive to less than ½ micro inch/inch.
External Circuits: Full Bridge or Half Bridge.
Accuracy: within 10 micro inch/inch of Bridge Output.
Power Supply: Integral battery. Sensitivity, Calibration and Null reading unaffected by battery decay.
Oscilloscope Jack: High impedance output.
Galvanometer Jack: Low impedance output (optional).

*PATENT PENDING

OTHER PRODUCTS:

Multi-Channel Strain Gage Switches
Switching and Balancing Units
Dynamic Strain Analyzer
Portable Torque Wrench Calibrator—Bolt Tester
Pressure Transducers—hydraulic & explosive
Load and Force Transducers
Weighing Systems

STRAINSERT BOLT†

(The Bolt With the Internal Strain Gage)

ACCURATE — CONTINUOUS — DIRECT INDICATION or RECORDING of

BOLT TENSION LOAD

Induced By:

- Tightening
- Vibration—Shock
- Test and Service Loads.

Connector Type C—Lead Wire Type W

†US Patent #2,873,341

Strainsert Company

the inside line of strain measuring devices

DIVISION OF POLYPHASE INSTRUMENT CO., BRIDGEPORT, PA.

Write Attention Alexander Yorgiadis for all Strainsert Products

Inquiries from Qualified Manufacturers' Representatives Solicited.

For details, circle No. 9 on Reader Information Card